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THE PROFESSION OF ENGINEERING

PRESIDENTIAL ADDRESS 1909

By President Jesse M. Smith, New York

Great engineering works existed in many parts of the world long before Columbus discovered America. We have but to consider the ruins left by the Incas in South America and the Aztecs in Mexico to realize the great work done on this continent in engineering. In Asia the great wall of China, the temples of Japan, China, Babylonia and Assyria bear record of the presence of the engineer.

2 In Africa, the vast pyramids of Egypt and the temples on the Nile are evidences that great engineers existed long before the Christian era. We marvel still when contemplating the pile of immense blocks of stone forming the pyramids and try to imagine what form of apparatus could have been used in placing those great stones one upon the other.

3 In Europe the Greeks and Romans did marvelous work in roads, bridges, aqueducts, and various mechanical structures which the modern engineer may well ponder upon and admire. While we read much in history of the emperors and kings who reigned when these great engineering works were produced, we learn little of the men who produced them, men whom we now call engineers.

4 While engineers have existed for thousands of years it is only within a comparatively recent time that they have begun to form themselves into societies for their mutual education and the advancement of the profession of engineering.

5 In England, as early as 1771, Smeaton and his contemporaries came together to form the Smeatonian Society of Engineers, which, therefore, according to the calculations of a noted English engineer, is five years older than the United States. The Institution of Civil Engineers of Great Britain came into existence in 1818, and was followed by its sister society, the Institution of Mechanical Engineers, in 1847. La Société des Ingénieurs Civils de France was founded in 1848. Die Verein Deutscher Ingenieure was organized in 1856.

Presented at the Annual Meeting of The American Society of Mechanical Engineers, December 1909.

6 In this country the Boston Society of Civil Engineers began its work in 1848. Our elder sister among national societies, the American Society of Civil Engineers, was organized in 1852. The next member of the family, the American Institute of Mining Engineers, was born in 1871. Our own Society came into existence in 1880, and our younger and very vigorous sister, the American Institute of Electrical Engineers, came along in 1884.

7 Each of these four national societies, the American Society of Civil Engineers, the American Institute of Mining Engineers, The American Society of Mechanical Engineers and the American Institute of Electrical Engineers, has grown greatly since its organization, and each continues to thrive. During the process of upbuilding of these four great national societies, several other national societies of specialists in engineering and many local societies of engineers have been formed, and all of these also are active and thriving.

8 The four greater national societies have an aggregate membership at this time of over 19,000 members. Twelve national societies of engineering specialists contain more than 13,000 members. Twenty-three local engineering societies in different cities of the United States count over 8,600 in their membership.

9 What does this great army of over 40,000 engineers, organized into many different societies, all for purely professional purposes, mean? It means that the engineering profession is making itself felt in this country of ours,—that it proposes to take a prominent place in the great activities by which the country is being developed,—that it will take its place in public affairs,—that it is coming into its own.

10 The national societies are not antagonistic to each other; on the contrary, they support and give confidence to each other. The national societies of specialists are not at war with the other national societies; they supplement them.

11 The local societies are not in opposition to the national societies; they extend their influence; they are the outposts of the great army. The specialists do not interfere with each other. We are all specialists to a greater or less extent; but we are all *engineers.*

12 In the legal profession, some men practice in the criminal courts: others devote themselves to titles in real estate; others are in corporation law; others practice in patent causes; they all squabble with each other in their practice; but when they meet in their bar associations they are all lawyers; they stand by each other and their profession; they are a power in the world.

13 The medical profession is made up of surgeons, oculists, aurists, general practitioners, specialists of the skin, the heart, the lungs and every other part of the human anatomy; but when they come together in their general medical associations they are all doctors; they also stand by each other and their profession; they also are a power in the world.

14 In the engineering profession why may not the men who practice in steam engineering; in machine construction; in hydraulics; in railroad, bridge, mining, electrical and chemical engineering; in metallurgy, refrigeration, heating and every other specialty in engineering, come together, stand by each other and their profession, become known as *engineers* and be a power in the world?

15 When, in 1889, the Institution of Civil Engineers of Great Britain invited the four national American societies of civil, mining mechanical and electrical engineers to visit it in London, there was inaugurated a spirit of friendship and coöperation in the engineering profession which has grown stronger and stronger as the years have passed. Following the visit in London, La Société des Ingénieurs Civils de France, in the same year, invited the American societies to Paris.

16 Those who were fortunate enough to participate in those memorable demonstrations of hospitality cannot fail to realize how greatly the seed of coöperation sown in that year has fructified.

17 In 1900 this Society was again invited by the Institution of Civil Engineers and the Institution of Mechanical Engineers to visit them in England, and again invited by the French society to visit it in Paris. Thus the spirit of coöperation was still further advanced by these remarkable meetings. On both occasions the sister societies abroad were untiring in the entertainment of the American engineers.

18 The year 1904 was made memorable by the acceptance of an invitation extended by this Society to the Institution of Mechanical Engineers of Great Britain to hold a joint meeting in Chicago. Thus the spirit of coöperation and good friendship was again strengthened and extended.

19 Now the Institution of Mechanical Engineers of Great Britain has expressed the desire to still further promote this friendly spirit by inviting this Society to a joint meeting in July of 1910 in England. The Council of our Society has accepted this very cordial invitation of the Institution in the spirit of good will in which it was extended. It remains for the membership of The American Society of Mechani-

cal Engineers to respond to this spirit and to go to England next year with its best talent and its best men.

20 The helpful coöperation in professional work which has already been established with our sister societies over the seas is also becoming manifest in our own country. The four national societies of civil, mining, mechanical and electrical engineers on March 24, 1909, held in this auditorium a joint meeting on the "Conservation of the National Resources," which did much to bring engineers close together and into coöperative relation.

21 Our Society invited the Boston Society of Civil Engineers to join in the monthly meetings of the Society recently held in Boston. The Engineers' Club of St. Louis in like manner was asked to join with us in the Society's monthly meetings recently held in St. Louis. In both cases the invitations have been accepted in the best spirit of coöperation.

22 The engineering societies of the country may be likened to the members of a large and harmonious family, each member independent to do its own special work in its own way, each member ready to help each of the others, each residing in its own home, but all ever ready to stand by each other, to work for the common good, to advance and dignify the profession of engineering.

23 A striking example of the "getting together" of the engineering societies is found in this building which is the home of our Society. It is also the home of our sister societies, the American Institute of Mining Engineers and the American Institute of Electrical Engineers.

24 Under the same roof are grouped together fifteen other societies of engineering and allied arts. 25,000 engineers practicing in all the specialties of engineering may call this building their professional home. We are living together here in peace and harmony. We have brought our books together into a single library open to the profession and to the public, where every person is welcome.

25 Our meetings are held in the same auditorium and lecture halls; the doors stand open that all who wish may enter. Our professional brethren of every society of every country are welcome here. The large hall at the entrance to the building is a foyer where all engineers may come together on the same plane, where they may unite to strengthen each other, to sustain and advance the profession of which they form a part.

26 The spirit of coöperation which now exists must be fostered, strengthened, made enduring, to the end that as great solidarity will

exist in the engineering profession as exists in any of the other great learned professions.

27 Numbers in membership are, of course, important in the societies which represent the engineering profession, but a high standard of membership is of much greater importance.

28 With a considerable number of high-grade technical schools throughout the country all striving with each other to raise the standards of engineering education ever higher and higher; and with the graduates from these institutions taking, from year to year, a larger and more responsible part in the great activities of the country, there is no lack of high-grade material from which to form a membership in the engineering societies which will be worthy of the profession.

29 In the Institution of Civil Engineers, as well as in the Institution of Mechanical Engineers, of Great Britain, we are informed, no person is admitted into the lower grade of membership unless he can pass a satisfactory examination as to the fundamental principles of engineering, by an examining board of the Institution. The rules laid down by this examining board form the standard by which the applicants to membership are measured. If the technical schools in Great Britain maintain an equally high standard in granting their degrees in engineering, then the degree may be accepted in lieu of an examination.

30 In other words, the engineering institutions in Great Britain establish the standard for the degrees granted by the technical schools. A promotion from a lower to a higher grade of membership is only made upon a showing of sufficient experience in engineering to satisfy the rules laid down by the Institution.

31 In The American Society of Mechanical Engineers, a person may enter the Society as a Junior upon the presentation of a degree in engineering from a technical school. But this Society has not, up to the present, established a standard by which to measure that degree. I believe the standard for such a degree in engineering should be established by the Society, and that it should be as high as that of the best schools of engineering in this country. It will follow that the schools having a lower standard will soon be brought up to the higher standard.

32 Promotion to higher grades of membership in our Society is only made upon a showing of engineering experience satisfactory to our Membership Committee. This committee is maintaining a high standard of membership, and I believe that acting under the influence of the membership and the Council of the Society, it will not allow that standard to fall, but rather cause it to rise.

33 If we are to have a profession of engineering, as distinguished from the trade of engineer, we must have a broad education befitting men of a learned profession, as distinguished from a narrower education sufficient for men of a trade.

34 President Lowell of Harvard in his recent remarkable inaugural address, gave this as his conclusion: "The best type of liberal education in our complex modern world aims at producing men who know a little of everything and *something well.*" If that conclusion be true of a liberal education leading to the learned profession of the law or medicine or theology, why is it not also true of a scientific education leading to the learned profession of engineering?

35 If preponderance be given to one part of President Lowell's conclusion over the other part, certainly knowing "a little of everything" leads to superficiality; while just as surely knowing but one thing well leads to narrowness. There would seem to be a happy mean between these two extremes, in the education of the engineer.

36 The engineer capable of being at the head of the larger engineering works must know something of many things, several things well and one thing profoundly.

37 The engineer president of a great railway system, for example, must know *something* of the alignment and gradients of the permanent way, its construction and maintenance; *something* of the proper location of sidings and stations; *something* of the system of signals, of the various kinds of cars, of the quality of water for the locomotives, of the heating and lighting of cars, and many other things. He must know *well* that the bridges have been designed for safety and endurance and that they have been properly constructed. He must know *well* that the tunnels are safely protected against external pressure and falling rocks. He must know *well* that the locomotives for drawing the high-speed trains, as well as those for the heavy freight trains, are of the very best design and capable of performing their duty with efficiency, economy and endurance. He must know *well* how to manage the traffic and keep the accounts. He must know *profoundly* how to coördinate all the different parts of this complex organization so that each part will perform its proper and full function, to the end that passengers and freight will be carried safely, surely, quickly and cheaply, and also that dividends will be paid to the shareholders.

38 The engineer knowing something of many things, several things well and one thing profoundly, is still one-sided if all this knowledge is confined strictly to his profession. He will be a much

broader man and a better engineer, if in his leisure hours he can turn his thoughts entirely away from his professional work and toward those things in nature and art which give that rest and renewal of the professional mind necessary to continued work.

39 Engineers have known for many years that the profession of engineering is a learned profession; the rest of the world is rapidly arriving at the same conclusion.

40 When in April, 1907, this building was dedicated "To the advancement of Engineering Arts and Sciences," President Hadley of Yale, where the learned professions have been taught for nearly 200 years, said:

> The men who did more than anything else to make the nineteenth century different from the other centuries that went before it, were its engineers.
>
> Down to the close of the eighteenth century the thinking of the country was dominated by its theologians, its jurists, and its physicians.
>
> These were by tradition the learned professions, the callings in which profound thought was needed, the occupations where successful men were venerated for their brains.
>
> It was reserved for the nineteenth century to recognize the dominance of abstract thought in a new field—the field of constructive effort—and to revere the trained scientific expert for what he had done in these lines.
>
> Engineering, which a hundred years ago was but a subordinate branch of the military art, has become, in the years which have since elapsed, a dominant factor in the intelligent practice of every art where power is to be applied with economy and intelligence.

It is encouraging to engineers to have their profession recognized as a "learned profession" by so great an authority as the president of Yale University.

41 Enthusiasm and devotion to his profession is characteristic of the engineer, and from my observation these begin with the student in engineering and extend right through his life. President Wilson of Princeton, in an address at Harvard not long since, dwelt upon "the chasm that has opened between college studies and college life. The instructors believe that the object of the college is study, many students fancy that it is mainly enjoyment, and the confusion of aims breeds irretrievable waste of opportunity." These conditions, I believe, exist to a much smaller extent in the technical schools, where engineers are taught, than in the general colleges, where a liberal education is obtained.

42 Enthusiastic love of work, for his profession's sake, resides in the heart of the engineer who becomes great. The man who merely works for wages, and without enthusiasm, does not rise; he remains a paid servant, and poorly paid at that.

43 Where enthusiasm exists, love of work exists; success follows. Our individual enthusiasm is quickened by the study of the work of our brother engineers.

44 What engineer while being whisked through the tunnels which connect Manhattan Island with the lands surrounding it, can fail to rejoice in his profession as he contemplates the work of the civil engineers, the mining engineers, the mechanical engineers, the electrical engineers, which, joined together, supplemented each other to produce success in those marvelous undertakings? The highest knowledge and skill in each of the four branches of the engineering profession were called for, and were forthcoming, in the consummation of this great work. It is not a question of which engineers did the most toward the success of this problem in transportation; they all did their best; they all did well; each contributed a necessary part to the success; they were all engineers working for the advancement of the profession of engineering.

45 Will not every true engineer feel his enthusiasm in his profession quicken, as he watches the great vessels of trade and the great vessels of war sweep out to sea, and he stops to consider how much of brains, and long experience, and hard work of many men are concentrated in each one of them?

46 We marvel still, our enthusiasm is inspired, as we see ponderous steam locomotives and mysterious electric locomotives competing in the hauling of trains, ever heavier and heavier, ever faster and faster, and both succeeding.

47 The automobile in its present highly developed and thoroughly practical form is the result of enthusiastic work of many engineers principally within the last fifteen years.

48 The enthusiasm of the engineer is never satisfied. Having conquered the highway with the automobile driven by the internal combustion gas engine, he now proposes to conquer the air with the aëroplane driven by the same kind of an engine in improved form.

49 The American Society of Mechanical Engineers has before it a future of usefulness to its members and influence in the profession, which is unlimited. It only requires that we stand by our tradition of increasing the membership with men of high quality as engineers; that the members maintain enthusiastic devotion to good professional work; that they coöperate with each other in the broadest and most friendly spirit to produce that solidarity of membership, and devotion to high ideals, which will compel the world to class the profession of engineering with the other learned professions.

EXPERIMENTAL ANALYSIS OF A FRICTION CLUTCH-COUPLING

By Prof. Wm. T. Magruder, Columbus, O.
Member of the Society

The following series of experiments was recently made to determine the results from the application of a known force at the end of the shifter lever of a friction clutch-coupling. Several 24-in., four-jaw friction clutch-couplings were used. They were the stock couplings made by the Falls Rivet & Machine Company, of Cuyahoga Falls, O. They consisted of the usual shifter lever, fork, yoke and cone, sliding on the driving shaft. The clutch arm *G*, Fig. 1, was a heavy casting keyed to the shaft *D*. Guide surfaces *H* were machined in each arm of the casting *G*, in which slid the inner jaws *I* and the outer jaws *J*. Each of the four pairs of jaws *I* and *J* was connected by pins *K* to the wedge-block *L*, which was fulcrumed at its center *M* in the clutch-arm casting. The wedge-blocks *L* carried adjustable steel wedges *N*, whose inner ends *O* engaged the short and hardened ends *P* of the cone-levers *Q*, and whose longer ends *R* were operated through the double links *F* by the sliding cone *E*. The inner and outer jaws engaged the annular ring *S* which was keyed to the driven shaft. This ring was 24 in. external diameter and 23 in. internal diameter. The eight jaws were each lined with a maple block $2\frac{1}{8}$ in. by 9 in. in size.

2 The tests included five lines of investigation:

First: To determine the forces required to throw in the shifter lever at different speeds when the clutch was in motion and when the clutch was at rest, and before and after the load had caused the clutch to slip on the ring.

Second: With different adjustments of the wedges, to determine the relation of the forces applied at the end of the shifter lever at different points in its motion and the cor-

All papers are subject to revision.

responding axial forces, to the forces caused thereby to be exerted by the clutch shoes upon the ring of the clutch pulley.

Third: To determine the frictional resistances between the clutch shoes and the ring of the clutch-coupling, in motion and at rest under different loads.

Fourth: To determine the power transmitted for different adjustments of the wedges corresponding to different forces required to throw in the shifter lever, including the

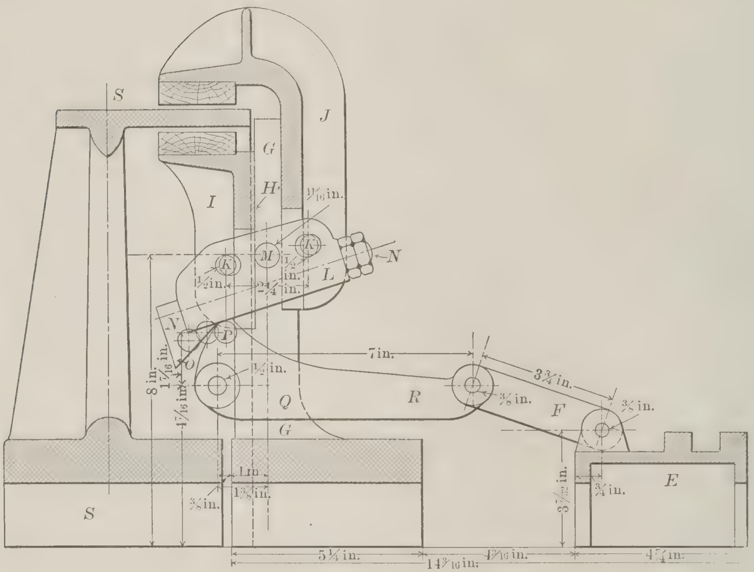

Fig. 1 Half Section of Friction Clutch-Coupling.

maximum power which the clutch was capable of transmitting, and the maximum power which it was capable of picking up from rest.

Fifth: To determine the relation of the maximum forces applied at the end of the shifter lever, and the corresponding axial forces, to the maximum power transmitted by two-arm and four-arm clutches for the same adjustment of the wedges.

3 To make these tests, two machines were designed and constructed, as follows:

DYNAMIC CLUTCH-TESTING MACHINE

4 To determine the maximum power which a clutch was capable of transmitting when the load was either gradually applied or picked up from rest, the dynamic clutch-testing machine, Fig. 2, was used. It consisted of floor-stands U, two co-axial shafts D and T, $3\frac{7}{16}$ in. in diameter, and a large belt pulley V on the end of the driving shaft D, to which power was delivered. The clutch-coupling G was placed near the middle at the junction of the two shafts. The coupling was keyed to the driving shaft D, and the clutch-ring S was keyed to the driven shaft T. On the opposite end of the driven shaft the brake pulley W was keyed. The shifter-lever A was operated in a horizontal plane. The motions required to throw in the clutch-cone E were measured. The lever A was operated by hand power, or by screw power B, from behind a screen Y made of planks and used for the protection of the persons engaged in the test. There was a horizontal slit in it for the motion of the lever.

5 To measure the force exerted on the end of the shifter lever, a calibrated spring-balance X was used. It was of 300 lb. capacity, and was graduated by 5-lb. divisions.

6 The power was absorbed by a prony brake Z from the internally flanged, flat-faced pulley W, 48 in. in diameter, and 24 in. face. The length of the brake arm Z was $72\frac{5}{32}$ in. The brake constant was 0.001145 b.h.p. per lb. per revolution. The effort exerted by the brake beam was measured by a platform scale C of 2000 lb. capacity. The leverage of the shifter lever A, when normal to the shaft D, was 4.939 in the dynamic clutch-testing machine. It is to be regretted that the power available was not sufficient to keep the speed uniform at 100 r.p.m., and for this reason the machinery slowed down to 92 r.p.m. under the heaviest loads.

STATIC CLUTCH-TESTING APPARATUS

7 In order to determine the force with which the shoes pressed against the clutch ring when a given maximum force was required to throw the shifter lever into operation, a static clutch-testing apparatus, Fig. 3 and Fig. 4, was used. It consisted of a clutch arm G mounted on a vertical shaft D supported in a flange coupling fixed to a structural steel frame a. Only the two opposite arms of the clutch were used in this apparatus. The inner jaws were removed. The shoes of the two outer jaws J were caused to press upon a dummy

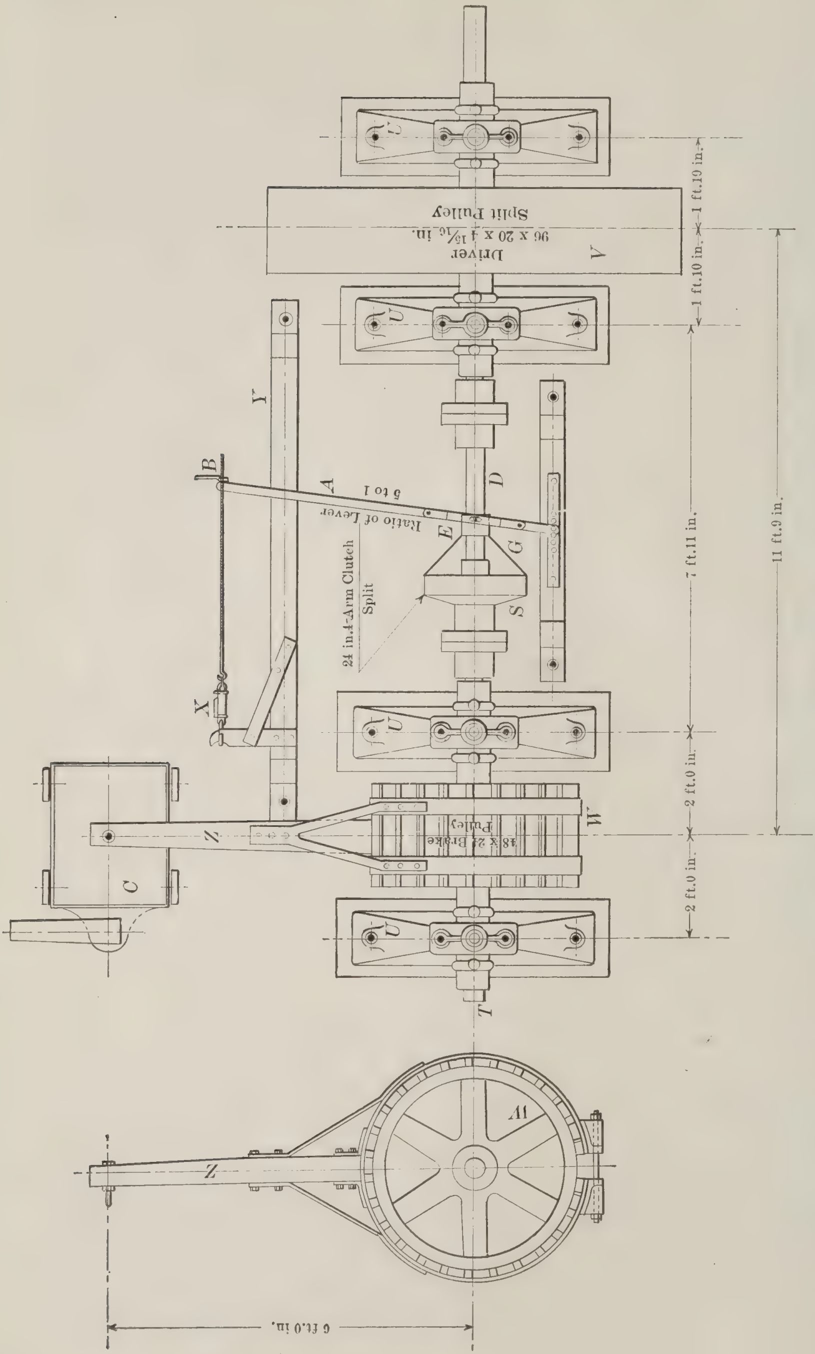

FIG. 2 DYNAMIC CLUTCH-TESTING MACHINE

ring made up of two separate cast-iron segments *b*. Each of these segments was connected by the links *c* to the yoke *d*, which in turn was connected through turnbuckle *e* to an eye carrying double knife edges *f*, which engaged the short and vertical end *g* of a bell-crank lever fulcrumed on a knife edge *h* in the frame *a* of the apparatus. To the longer and horizontal arm *i* of the bell-crank lever was knuckled a vertical prop *j*, the lower end of which was conical and which bore in a center-punch mark made in an iron bar resting upon a platform scale *k*, of 600 lb. capacity. The ratio of the arms was seven. Two platform scales were used, one for each dummy ring segment.

8 The parts of the mechanism were adjusted so that the knife edges were kept in position and bore fairly. The ring segments were so located that their external diameter was 24 in., or the same as the clutch ring. This was done by means of the turnbuckles. When the ring segments were in this position the wedges were adjusted to make the maple shoes bear evenly. The shifter lever was thrown in by screw power *B* and the force so required was measured frequently at definite intervals which were determined by measuring the distances which the cone *E* had been moved from its original position. The leverage of the shifter lever *A*, when normal to the shaft, was 4.863 to 1 in this apparatus.

9 In using the dynamic clutch-testing machine, the maple shoes were first adjusted by means of the wedge nuts so that they bore evenly. They were then burned in by driving the shaft *D* and the coupling *G* while the ring *S* was prevented from rotating. The wedge nuts were then adjusted again to make the shoes of both outer and inner jaws bear evenly on the ring when the shifter lever was thrown in.

10 This adjustment was tested by means of 16 copper strips, $\frac{1}{2}$ in. wide, 0.002 in. in thickness, one used at each end of each shoe. If the shoes did not bear evenly, or at least as well as they are supposed usually to do in ordinary good millwright's practice, the wedge nuts were screwed up, the ring blocked from rotating, the lever thrown in, and the shoes again burned in. By this means fairly uniform results and even pressures were obtained between the eight shoes and the ring. With the shifter lever thrown in and the copper strips just capable of being pulled out by hand, the counting of the rotations of the wedge nuts was begun. Similar adjustments were made on the static clutch-testing apparatus, except that the shoes had been surfaced but not burned in.

11 In the tests, the wedge nuts were all screwed up, one turn or

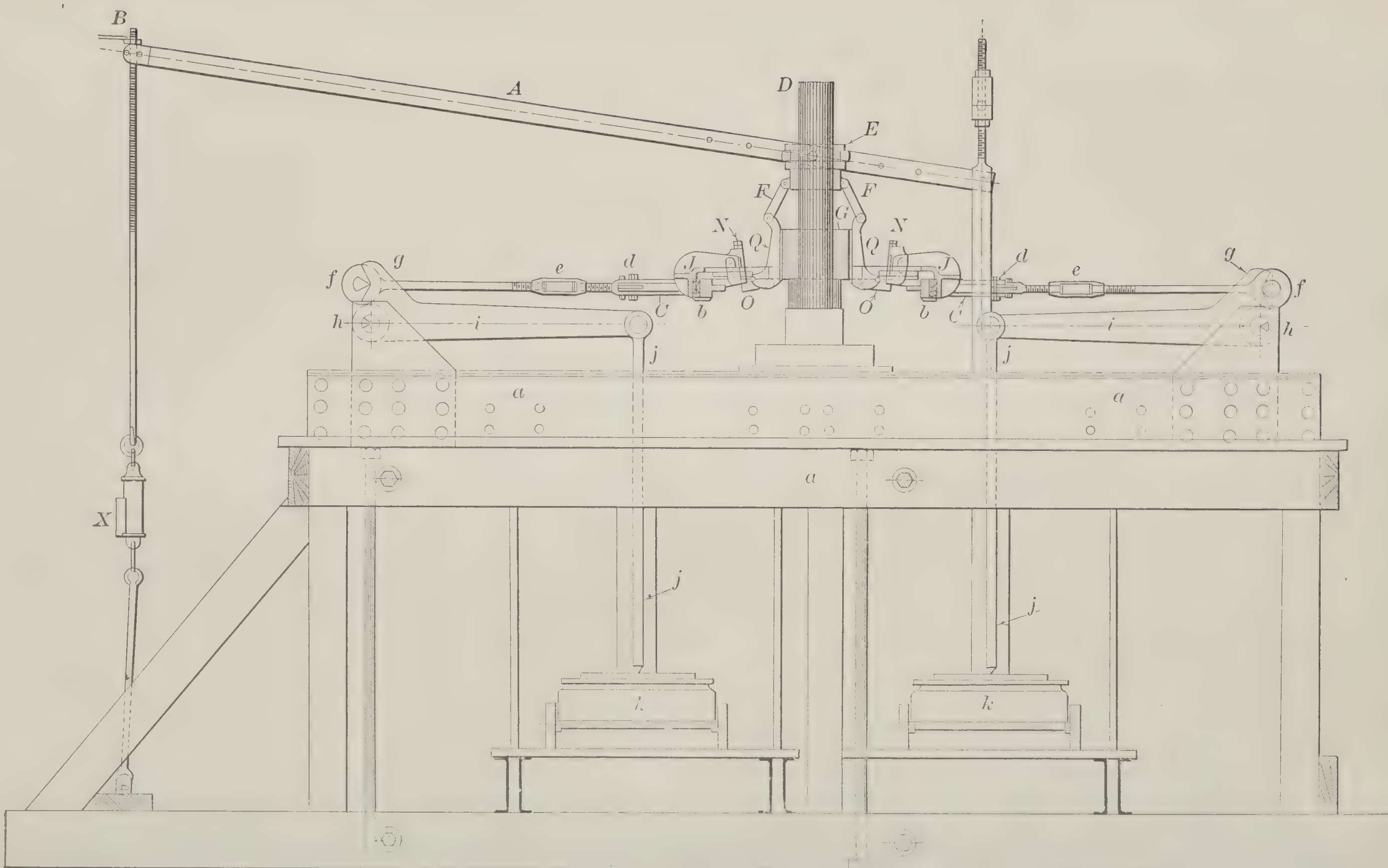

FIG. 3 STATIC CLUTCH-TESTING APPARATUS

less at a time, and the tests made, then another turn, and so on. When the shifter lever A was thrown in by hand power, a man applied his muscular effort to the spring balance X on the end of the lever: when it was thrown in by screw power, the tail-nut B was rotated. The spring balance had a maximum indicator besides the usual one. The motions of the cone E of the different couplings tested varied from 4 in. to $4\frac{1}{2}$ in. Readings of the spring balance were usually taken for each $\frac{1}{4}$-in. or $\frac{1}{2}$-in. motion of the cone.

FIRST SERIES OF TESTS

12 These were made to determine the forces required to throw in the shifter lever at different speeds, when the clutch was in motion and when the clutch was at rest, and before and after the load had caused the clutch to slip on the ring.

13 Tests G, H, I, and M were made on the dynamic clutch-testing machine. The forces required to throw in the shifter lever by screw power and by hand power when the shafts were at rest were first determined, the machine then started up, and the brake tightened until the clutch-coupling slipped on its ring. The brake was then loosened and another test made with the result that less power was transmitted. The wedge nuts were then tightened and the forces required to throw in the shifter lever were measured one or more times and the test continued, as given in Tables 1 and 1-A. In Test M, Table 1-A, readings of the forces required to throw in the shifter lever were taken both when the shafts were at rest and when they were in motion. The shifter lever was thrown in several times and the power determined for a fixed setting of the brake nuts. These were then tightened, and another set of four or more readings taken of the force required to throw in the shifter lever by hand power when the shaft was in motion.

14 With the wedge nuts screwed up two and one-half turns it required a maximum of 70 lb. to throw in the shifter lever by screw power. Immediately thereafter it required maxima of 55 lb. on the first trial, 45 lb. on the second trial, and 43 lb. on the third trial, to throw in the shifter lever by a steady pull by hand power. This shows that the force required to throw in the shifter lever by hand power was much less than by screw power. While this was partly due to the friction of rest being greater than the friction of motion, it was also partly due to the various parts of the clutch adjusting themselves to the conditions after one or two engagements of the

Fig. 4. General View of the Static Clutch-Testing Apparatus

shoes with the ring. This has been frequently noted in practice in the shop.

15 With the same adjustment of the wedges, the clutch slipped when transmitting 84.6, 71.1, and 65.0 h.p., on the first, second, and third sets of trials. This reduction in the brake load shows the effect of wear of the clutch shoes due to the slipping on the ring. This was also indicated by the fact that it required a maximum of only 31 lb. to throw in the shifter lever by hand power after the test.

TABLE 1 FORCES REQUIRED TO THROW IN SHIFTER-LEVER AND HORSE-POWER TRANSMITTED

Dynamic Clutch-Testing Machine

Wedges Set Up Turns	Distance From Start	Max. Force Required to Throw in Lever with Shaft at Rest. By			Corresponding Axial Pressure. Shaft at Rest. By		Gradually Applied Loads		
		Screw Power	Hand Power	Ratio	Screw Power	Hand Power	Net Brake Load	Rev. per Min.	Brake Horse-Power
		Tests G. 24-in., Four-Arm, Solid Clutch-Coupling.							
1.5	2.75	56			277		581	96	63.9
							497	96	54.6
							471	96	51.8
2.5			119			588			
			94			464			
			75			371			
3.0			92			454			
			75			371			
3.5			114			563			
			92			454			
			92			454			
3.5	2.5	103	91	0.883	509	450	1185	92	124.8
		87[1]			430[1]				
		Tests H. 24-in., Two-Arm, Solid Clutch-Coupling.							
3.5	2.5	45			222		471	96	51.8
							491	96	54.0
							486	95	52.9
4.0	1.875	114			563				
	2.25	111			548		735	95	79.9
		80[1]			395[1]				

[1] After test and after slipping.

TABLE 1—Continued

Wedges Set up Turns	Distance From Start	Max. Force Required to Throw in Lever with Shaft at Rest. By			Corresponding Axial Pressure. Shaft at Rest. By		Gradually Applied Loads		
		Screw Power	Hand Power	Ratio	Screw Power	Hand Power	Net Brake Load	Rev. per Min.	Brake Horse Power
Tests I. 24-in., Four-Arm, Split Clutch-Coupling.									
2.5	2.5	70	55	0.79	346	272	....	....	....
....	....	....	45	0.64	....	222	....	....	....
....	....	....	43	0.61	....	212	739	100	84.6
....	....	....	....	....	....	....	621	100	71.1
....	....	....	....	....	....	....	568	100	65.0
....	....	....	31[2]	0.44[2]	....	153[2]	....	....	....
3.5	....	85	65	0.76	420	321	....	....	....
....	....	....	67	0.788	....	331	915	96	100.6
....	....	....	55	0.646	....	272	....	....	....
....	....	....	53	0.623	....	362	....	....	....
4.5	2.25	133	99	0.74	657	489	....	....	....
....	....	....	100	0.75	....	494	1216	94	130.9
4.5	2.00	179	139	0.78	884	689	....	....	....
....	....	....	119	0.67	....	588	....	....	....
....	....	....	119	0.67	....	588	....	....	....
....	....	....	117	0.65	....	578	1371	93	146.0

[2] After test and after slipping. Reduction in b.h.p. shows effect of wear of shoes.

16 After two more sets of trials with tighter adjustments of the wedges, 130.9 h.p. was transmitted at 94 r.p.m.

17 With the wedge nuts screwed up four and one-half turns, it required a maximum of 179 lb. to throw in the shifter lever by screw power, and of 119 lb. to throw it in by a steady pull by hand, corresponding to 884 lb. and 588 lb. respectively, of axial thrust. Under these conditions, when transmitting 146 h.p. at 93 r.p.m., the split clutch broke. This is 77.7 per cent of the breaking load carried by the four-arm solid clutch.

18 From Tests G and I, Table 1, it will be seen that the force required to throw the shifter lever in by a steady pull by hand power, the first time, varied from a minimum of 66.5 per cent to a maximum of 88.3 per cent, averaging about 76.8 per cent of the force required to throw in the shifter lever by screw power; and that the force required to throw in the shifter lever by hand power, after the test,

TABLE 1-A FORCES REQUIRED TO THROW IN SHIFTER-LEVER AND PICKUP LOAD

Wedges Set Up Turns	Max. Force Required to Throw in Shifter-Lever. Shaft at Rest: Screw Power	at Rest: Hand Power	in Motion: Hand Power	in Motion: Ratio	Number of Trials Averaged.	Corresponding Axial Pressure. Shaft at Rest: Screw Power	at Rest: Hand Power	in Motion: Hand Power	Pick-up Loads: Actual Net Brake Load	Estimated rev. per Min.	Equivalent Brake Horse-Power
Tests M., 24-in. Four-Arm, Solid Clutch-Coupling.											
2.0	67				1	331					
		61		0.91	3		301				
			58	0.87	5			258	94	102	11.0
			44	0.67	4			217	206	100	23.6
3.0		124			4		612				
			93		5			459			
			86		4			425			
			69		4			341	543	96	59.7
4.0		140			1		692				
		125			6		617				
			115		4			568			
			105		5			519	501	96	55.1

varied from 44.3 per cent to 64.6 per cent, averaging 54.5 per cent of the force required to throw in the shifter lever by screw power before the test. From Tests I, the ratio of the forces required to throw in the shifter lever by hand power, before and after the tests, varied from 31/45 = 68.8 per cent to 54/66 = 81.8 per cent. It is to be noted that when the four-arm solid clutch broke, when transmitting 187.8 h.p., its wedges had been adjusted so that it required a maximum of 114 lb. to throw in the shifter lever by screw power, corresponding to a maximum of 563 lb. axial thrust. With the same axial thrust applied, the 24-in. two-arm clutch slipped when transmitting 79.9 h.p., or only 42.5 per cent thereof.

19 Tests L, Table 2, were made on the static clutch-testing apparatus. They give the forces required to throw in the shifter lever by screw power and by hand power. The ratios are higher than those given in Table 1 because there was no intermediate starting up, slipping and wearing of the clutch shoes. The last two tests were made with only one jaw in service, the other being disconnected.

20 From these tests it will be seen that the force required to throw in the shifter lever slowly and steadily by hand power averages on the static apparatus 88 per cent and on the dynamic machine about 79 per cent of that required to throw it in by screw power;

TABLE 2 TESTS L: FORCES REQUIRED TO THROW IN SHIFTER-LEVER, AND PRESSURES EXERTED BY CLUTCH-SHOES ON CLUTCH-RING

TESTS ON STATIC CLUTCH-TESTING APPARATUS. 24-IN., FOUR-ARM, SOLID CLUTCH-COUPLING, USING ONLY TWO OUTER JAWS

Wedges Set Up, Turns	Distance from Start	Max. Force Required to Throw in Shifter-Lever Shaft at Rest by			Corresponding Axial Pressure Shaft at Rest by		Net Platform Scale Readings Corresponding to				Actual Shoe Pressures Corresponding to Maximum Force. Lever Ratio = 7					Ratio Column 16 to Column 3
							Column 3		Max. Net		To Throw in Lever		On Shoes		Total on Shoes	
		Screw Power	Hand Power	Ratio	Screw Power	Hand Power	W	E	W	E	W	E	W	E		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
3	$1\frac{5}{16}$	44	....	.....	214	....	150	160	263	271	1050	1120	1841	1897	3738	85
3	$1\frac{9}{16}$	44	....	.....	214	....	173	182	....	....	1211	1274	.....	.....	.....	.....
4	$1\frac{1}{16}$	59	51	0.86	287	246	179	192	330	339	1253	1344	2310	2373	4683	79
5	$\frac{13}{16}$	77	68	0.86	374	331	207	225	400	409	1449	1575	2800	2863	5663	74
6	$\frac{13}{16}$	85	77	0.91	413	374	226	234	417	419	1582	1638	2919	2933	5852	69
6.5	$\frac{9}{16}$	97	90	0.92	472	435	203	216	453	460	1421	1512	3171	3220	6391	66
7	$\frac{5}{16}$	134	109	0.81	652	530	227	252	535	544	1589	1764	3745	3808	7553	56
7	$\frac{9}{16}$	110	100	0.91	535	486	249	261	500	504	1743	1827	3500	3528	7028	64
8	$\frac{5}{16}$	157	134	0.86	763	652	289	310	594	600	2023	2170	4158	4200	8358	53
Average				0.88	....	....	....	......	....	....	.....	.....	.....	.....	.....	68
4	$\frac{13}{16}$	33	29	0.88	160	141	125	out of	277	....	875	.....	1939	.....	.....	.....
6.5	$\frac{9}{16}$	52	48	0.92	253	233	180	service	394	....	1260	...	2758	.....	.....	.....

Maximum force required to throw in shifter-lever acting on outer jaw of only one arm:

By screw-power.. 33 lb.
By hand-power, steady pull.. 29 lb.
By hand-power, rapidly.. 31 lb.
By hand-power, more rapidly....................................... 37 lb.
By hand-power, suddenly, or by jerk............................... 55 lb.

that this ratio is reduced to about 0.68 by repeated trials of the mechanism when at rest; that the ratio of the forces required to throw in the shifter lever when the shafts are in motion and are at rest varies from 0.91 to 0.87 according to the number of slippings of the clutch on its ring, and may be reduced even to 0.67 after several slippings.

SECOND SERIES OF TESTS

21 The series was made to determine with different adjustments of the wedges the relation of the forces applied at the end of the shifter lever at different points in its motion and the corresponding axial forces, to the forces caused thereby to be exerted by the clutch shoes upon the ring of the clutch-pulley.

TABLE 3 TESTS L: FORCES REQUIRED TO THROW IN SHIFTER-LEVER AND CORRESPONDING PRESSURES EXERTED ON PLATFORM SCALES

TESTS ON STATIC CLUTCH-TESTING APPARATUS. 24-IN., FOUR-ARM, SOLID CLUTCH-COUPLING, USING ONLY TWO OUTER JAWS

Distance Cone is Moved	Corrected Spring-Balance Readings of Forces Exerted at End of Shifter-Lever	Corresponding Net Platform Scale Readings	
Inches		West	East
$\frac{1}{16}$	18	8	12
$\frac{5}{16}$	43	55	65
$\frac{9}{16}$	53	102	116
$\frac{13}{16}$	57	144	159
$1\frac{1}{16}$	59	179	192*
$1\frac{5}{16}$	58	209	222
$1\frac{9}{16}$	53	235	248
$1\frac{13}{16}$	51	256	269
$2\frac{1}{16}$	48	271	287
$2\frac{5}{16}$	43	290	304
$2\frac{9}{16}$	38	303	314
$2\frac{13}{16}$	33	312	323
$3\frac{1}{16}$	32	318	330
$3\frac{5}{16}$	26	321	335
$3\frac{9}{16}$	23	326	337
$3\frac{13}{16}$	16	328	339
$4\frac{1}{16}$	8	330	339
$4\frac{3}{16}$	0	329	339†

* Maximum Force required to throw in shifter-lever.
† Maximum pressure on shoes.

22 Table 2 gives the results of the tests with the static apparatus. The maximum forces required to throw in the shifter lever by screw power and by hand power, and their ratio, the corresponding axial pressures, the corresponding net platform-scale readings, and the actual shoe-pressure readings for these two maxima, are given. "W" and "E" mean "west" and "east" and refer to the relative positions of the two scales on the left and right-hand sides respectively, as in Fig. 4.

23 Comparing the figures in the last column, which give the ratio of the sum of the maximum actual shoe pressures to the maximum force required to throw in the shifter lever, it is seen that the force ratio varies from 85 for three turns of the wedge nuts to 53 for eight turns of these nuts, with an average of 68. This shows that the efficiency is much greater under the lesser pressures.

24 The two tests at seven turns show the effect of compression in the form of set, not only on the shoes but on the various parts and joints of the clutch.

TABLE 4 TESTS F: FORCES REQUIRED TO THROW IN SHIFTER-LEVER, AND HORSEPOWER TRANSMITTED

TESTS ON DYNAMIC CLUTCH-TESTING MACHINE. 24-IN., FOUR-ARM, SOLID CLUTCH-COUPLING

Wedges Set Up, Turns	Distance From Start	Max. Force Required to Throw in Shifter-Lever. Shaft at Rest. By Screw-Power.	Corresponding Axial Pressure. Shaft at Rest. By Screw-Power.	Gradually Applied Loads. Net Brake Load	Gradually Applied Loads. Rev. per Min.	Gradually Applied Loads. Brake Horse-Power
	$2\frac{7}{8}''$	51	252	556	90	57.3
	......	....	....	666	94	71.7
	......	....	....	606	97	67.3
One more turn.........	$2\frac{5}{8}''$	97	479	1112	92	117.1
One more turn.........	$2\frac{5}{8}''$	114	563	1665	94	179.2
				1745	94	187.8
Equivalent horsepower at 100 r.p.m.					100	199.8

25 The following statements are deduced from the results of this series of tests:

a The average ratio of the maximum forces required to throw in the shifter lever with one and two jaws was 0.55.

b The average ratio of the corresponding forces exerted on the ring with one and two outer jaws, counting the forces exerted by only one jaw in each case, was 0.79.

c The average ratio of the corresponding forces exerted on the ring with one and two outer jaws, counting the forces exerted by both the jaws, was 0.38.

d The average ratio of the maximum forces exerted on the ring with one and two outer jaws, counting the forces exerted by only one jaw in each case, was 0.85.

e The average ratio of the maximum forces exerted on the ring with one and two outer jaws, counting the forces exerted by both the jaws, was 0.42.

f In other words, 55 per cent of the force applied at the shifter lever produced only 38 per cent as much corresponding force exerted on the ring, and only 42 per cent as much of the maximum force exerted on the ring, for one jaw rather than two jaws. This was doubtless due to the inequality of the pressures exerted when only one arm was in use, and shows the desirability of so adjusting the wedges of the opposite arms that the shoes bear equally.

26 Table 3 shows the relation of the motion of the cone to the maximum force required to be applied at the end of the shifter lever and to the forces exerted by the clutch shoes upon the dummy ring-segments in Tests L, and is a fair sample showing that the maximum force exerted on the shifter lever does not produce the maximum force exerted on the clutch shoes.

TABLE 5 TESTS N: FORCES REQUIRED TO THROW IN SHIFTER-LEVER, AND HORSEPOWER TRANSMITTED WITH GRADUALLY APPLIED AND PICKED-UP LOADS

TESTS ON DYNAMIC CLUTCH-TESTING MACHINE. 24-IN., FOUR-ARM, SOLID CLUTCH-COUPLING.

Wedges Set Up, Turns	Distance From Start	Maximum Force Required to Throw in Lever. Shaft at Rest. By Screw-Power.	Corresponding Axial Pressure Shaft at Rest. By Screw-Power	Gradually Applied			Suddenly Applied Loads		
				Net Brake Load	Rev. per Min.	Brake Horse-Power	Net Brake Load	Rev. per Min.	Brake Horse Power-
3	2.5	53	263	711	96	78.2	471	96	51.8
3	2.75	43	212[1]						
4	2.25	90	445	1035	92	109.0	471	98	52.9
4	2.5	58	286[1]						
5	2.25	125	617	606	96	66.6	491	98	55.1
				606	96	66.6	561	96	61.7
				746	94	80.5	628	95	68.3
				621	93	66.1	981	91	102.2
							986	91	102.7
Equivalent horsepower at 100 r.p.m.								100	112.8

[1] After slipping three times.

THIRD SERIES OF TESTS

27 The third series of tests was made to determine the frictional resistance between the clutch shoes and the ring of the clutch-coupling in motion and at rest.

28 For these tests, a cast-iron plate, $1\frac{15}{16}$ in. thick, 12 in. wide, and 36 in. long, and maple blocks $\frac{27}{32}$ in. thick, 3 in. wide, and 9 in. long were used. The angles of inclination of the plate at which a block would begin to slide from rest, and at which it would continue to slide after being started into motion, were taken as the angles of friction respectively for the two cases. The horizontal forces required to be exerted in order to start the block from rest, and to continue it in motion when placed on the carefully leveled plate, were taken to be the natural tangents of the angles of friction respectively for the weights carried by the block.

29 The smoothness of finish of the block, the uniformity and trueness of the bearing surface, the deflection of the plate, the cushion of air between the block and the plate, each has its effect on the angle of friction.

30 The number of tests made with flat maple blocks does not warrant the drawing of very positive conclusions, but it would seem that the average frictional resistance under load was greater from rest than the resistance under load in motion, in the proportion of the tangent of 18.5 deg. to the tangent of 13.3 deg., or in the proportion of 0.33 to 0.24.

FOURTH SERIES OF TESTS

31 The next series of tests was to determine the power transmitted for different adjustments of the wedges corresponding to different forces required to throw in the shifter lever, including the maximum power which the clutch-coupling was capable of transmitting, and the maximum power which it was capable of picking up from rest.

32 For these tests, the dynamic clutch-testing machine was used. The wedges were adjusted so that the shoes bore fairly equally. To determine the maximum power which the clutch-coupling was capable of transmitting, the wedge nuts were gradually tightened, and the brake screwed up either until the coupling slipped, in which case the wedge nuts were tightened up further, or else the clutch broke. Table 4 gives the results of this set of tests, from which it will be seen that with a maximum of 114 lb. applied by screw power at the end of the shifter lever, corresponding to an axial thrust of 563 lb., when revolving at 94 r.p.m. under a net brake load of 1745 lb. the clutch transmitted 187.8 h.p., under which condition the clutch slipped, the speed varying from 94 to 98 r.p.m., and both clutch and ring broke. This corresponds to 199.8 b.h.p., or practically to a maximum of 200 b.h.p., for a speed of 100 r.p.m.

33 Table 5 gives the results of the tests with the dynamic clutch-testing machine, of the forces required to throw in the shifter lever, and the horsepowers transmitted with gradually applied and suddenly applied loads. The latter are what are sometimes called pick-up loads. From this table it will be seen that with a net brake load of 986 lb., when running at 91 r.p.m., the clutch picked up 102.7 h.p. and had it started when the clutch broke. It had just previously picked up 102.2 h.p. This corresponds to 112.8 maximum b.h.p. of pick-up load for a speed of 100 r.p.m.

FIFTH SERIES OF TESTS

34 The last series was to determine the relation of the maximum forces applied at the end of the shifter lever and the corresponding axial forces, to the maximum power transmitted by two-arm and four-arm clutches for the same adjustment of the wedges.

35 To perform this test on the clutch which had been tested in the dynamic clutch-testing machine (See Tests H of Table 1), the two opposite pairs of jaws were disengaged by unscrewing their wedge nuts, and retaining the same adjustment on the two other pairs of shoes. It was found that it then required a maximum of 45 lb. to throw the shifter lever in by screw power, and that when revolving at 95 to 96 r.p.m. the clutch slipped at 51.8, 54.0, and 52.9 b.h.p. respectively. When the wedges were tightened up one half turn further, it required a maximum of 114 lb. to throw the shifter lever in by screw power. When running at 95 r.p.m. the clutch slipped when transmitting 79.9 h.p. After the test, it required a maximum of only 80 lb. to throw the shifter lever in by screw power.

36 Comparing the tests of these clutches, with four arms and two arms, the wedge nuts being turned up three and one-half turns in both cases, the horsepowers required to slip the clutch were found to be 124.8 and 52.9 (average of 51.8, 54.0, 52.9). This would seem to show that the two-arm clutch transmitted only 44 per cent as much power as would the same clutch with four arms for the same adjustment of the wedges. As the axial thrusts, however, were in the same proportion, 103 to 45, it would seem as though the horsepowers transmitted were directly proportional to the number of arms, whether two or four, and to the forces required to throw the shifter lever in by screw power, and therefore to the axial thrusts. No tests were made with six-arm clutches.

CONCLUSIONS

37 Applying the deductions of Tests L, Table 2, to Tests F, Table 4, we may say

A That at 100 r.p.m., with the shoes properly burned in and the wedges adjusted so as to give equal pressures between each of the eight shoes and the ring, and with no excessive lost motion between the jaws and their guides in the clutch-arm casting, a 24-in. four-arm solid clutch and ring will probably break:

a When transmitting 200 h.p., if gradually applied.

b When attempting to pick up a load exceeding 110 h.p.

B That to do so will require:

a A maximum force of between 100 lb. and 115 lb., applied at the shifter lever for a leverage of five.

b A maximum axial force or thrust on the collar of between 500 lb. and 600 lb.

c A combined maximum pressure of the eight shoes on the clutch ring of between 7500 lb. and 8000 lb.

d An intensity of pressure of about 50 lb. per sq. in. for each of the twenty square inches of each of the eight shoes of the four-arm clutch.

C That any inequality or lack of evenness and uniformity of the pressures with which opposite shoes bear on the ring, or any lost motion between the various parts, will decrease the breaking strength of the clutch.

D That a 24-in. four-arm split clutch will probably break when transmitting between 140 and 150 b.h.p. at 100 r.p.m., if the force is gradually applied and under proper conditions.

E That the factor of safety of 10, as used by the clutch manufacturer in this case, is quite ample.

AN ELECTRIC GAS METER

By Prof. C. C. Thomas, Published in The Journal for December

The following addition to his paper was given orally by Professor Thomas in presenting it before the Society at the meeting of December and should therefore be considered a part of the paper.—Editor.

THEORY OF THE METER AND METHOD OF OBTAINING STANDARD RESULTS

32 The figures given in paragraphs 14, 15 and 19 can be reduced to standard conditions of temperature and pressure, and the meter readings can be autographically recorded directly in "standard cubic feet" of gas or air. Let

G = cubic feet of gas per hour
E = energy in kilowatts

Then B.t.u. per hr. = 3412 E

T = temperature difference, deg. fahr.
S = specific heat per cu. ft.

Then $G\ S\ T$ = heat energy equivalent to E, or $GST = 3412\ E$. $\frac{GT}{E} = \frac{3412}{S}$ = a constant K which depends upon the specific heat of the gas.

33 Since the temperature difference T is kept constant, it follows that $\frac{K}{T}$ is constant. Let $\frac{K}{T} = C$. Then $G = \frac{KE}{T} = CE$.

34 It is now proposed to show by reference to the gas and the air curves in Fig. 10, that if the specific heat of gas made under given conditions be calculated from the customary chemical analysis and the specific heat of the constituents, then this specific heat may be used for determining the constant C. From the gas curve (Fig. 10), which was made with illuminating gas at an average temperature of

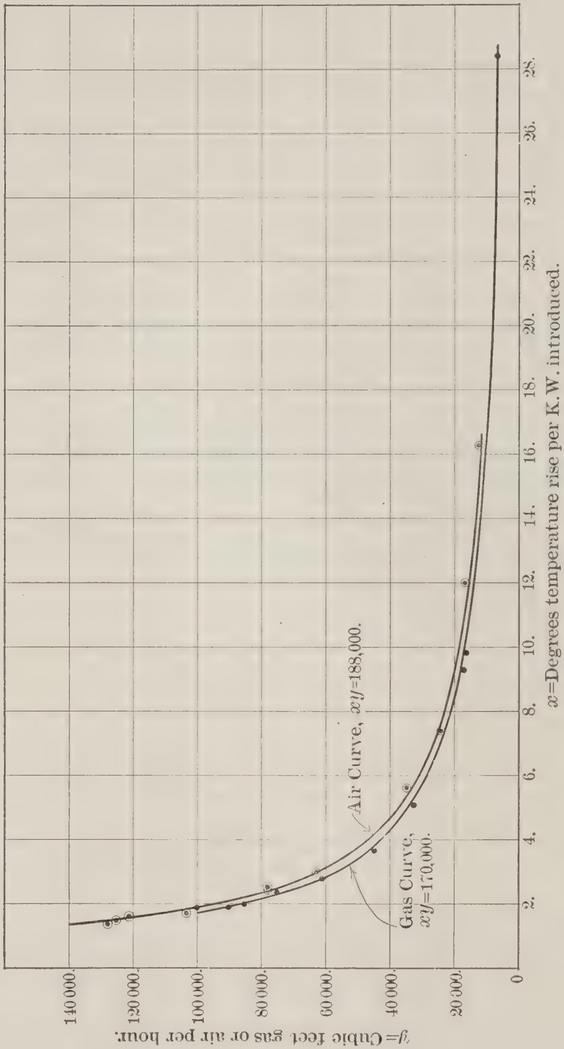

FIG. 10 SHOWING DEGREES TEMPERATURE RISE PER KW. FOR DIFFERENT RATES OF FLOW OF GAS AND AIR

59 deg. fahr., and under an average absolute pressure of 6 in. water and 29.8 in. mercury,

$$K = 170{,}000 = \frac{3412}{S}$$

35 Therefore for the condition of the gas when the tests were made the specific heat per cubic foot $= S \; \frac{3412}{170{,}000} = 0.0201$. If this be reduced to standard conditions of 32 deg. fahr. and 29.9 in. mercury, then $S = 0.021$, which is to be compared with the calculated specific heat (Par. 14), giving $S = 0.0211$. If the standard conditions are taken as 62 deg. fahr. and 29.9 in. mercury, the specific heat becomes 0.0198, and the constant becomes

$$K = \frac{3412}{0.0198} = 172{,}500, \text{ nearly}$$

If the temperature difference is kept constant at 5 deg. fahr., then $\frac{K}{T} = \frac{172{,}500}{5} = 3450 = C$, or $G = 3450\ E$.

36 The cross-section paper on the recording wattmeter is ruled so that 3450 E is read directly, instead of the watts E. The record is thus read directly in cubic feet of gas. The regular records of chemical analysis of the gas should be referred to from time to time in order to ascertain what percentage variation takes place in specific heat. It appears, as stated previously, that the elements which vary during the operation of a gas plant are not those whose variation would produce serious variation in specific heat. The variation that does take place is apparently well within the limits of accuracy practicable, or generally considered necessary in the operation of gas plants. By taking frequent chemical analyses the error can be reduced so as to be quite negligible.

37 The conditions during the air tests were as follows: barometer, 29.75; pressure, 6.5 in. water; average temperature of air as measured in the wet meter, 60 deg. fahr. From the air curve obtained under these conditions (Fig. 10)

$$K = 188{,}000, \text{ and } S = \frac{3412}{188{,}000} = 0.0181$$

38 Reducing this to standard conditions of 32 deg. and 29.9 in. mercury, $S = 0.0191$. This is to be compared with the accepted specific heat of air under these conditions, or 0.0192 B.t.u. per cu. ft.

This provides perhaps the best evidence that could be obtained, as to the accuracy of these tests, since the specific heat of air is well known at the conditions under which the tests were made. A more commonly familiar figure for specific heat of air is obtained by multiplying 0.0192 by the number of cubic feet of air per pound under the above conditions, or 12.38. The result is 0.2377 B.t.u. per lb. per deg. and this is to be compared with 0.0191×12.38 as given by the meter, or 0.2365.

39 The constant K for air at 32 deg. and 29.9 in. is therefore

$$\frac{3412}{0.0191} = 178{,}630$$

and reducing this to 62 deg. instead of 32 deg.

$$K = \left(1 \times \frac{30}{493}\right) \times 178{,}630 = 189{,}500 \text{ nearly}$$

If $T' = 5$ deg., $\frac{K}{T} = 3790$.

40 The error involved in calling this constant 3800 is less than $\frac{1}{2}$ of 1 per cent and well within the limits of accuracy possible under the circumstances. The standard cubic feet of air passing the meter are therefore $G = 3800\ E$, and the autographic records are arranged to read accordingly, in standard cubic feet of air per hour.

41 The development of a new device requires consideration of a large number of questions arising out of the conditions of service proposed. The question of specific heat has been considered in the preceding paragraphs. The degree of success which has been attained with this meter in accurately measuring specific heat is due principally to an extensive experience in this particular class of work, which has served to point out the way to make an electrical heater in which heat losses are negligibly small. The arrangement of the meter is such that the heat given off can go into the gas only, and it necessarily all goes into the gas, with the exception of a negligibly small loss which it is not worth while to minimize further. That the gas receives all the heat, excepting this negligibly small loss, is true whether or not the heating material has collected deposit of some kind. So long as the gas can get through the heater, its temperature is raised proportionately to the heat supplied.

42 The question of the presence of a small amount of water vapor, as part of the gas, has so far not introduced any complications. It

is conceivable that if the gas carried a large percentage of water the operation of the meter would be interfered with,—but so would the operation of a gas engine or a burner. The meter can apparently measure accurately any gas that can be used by a gas engine. The absence of moving parts in the meter gives it an advantage over the engine, and dust can be to a considerable extent deposited before entrance of the gas to the meter. The heating element and thermometers can be cleaned by dipping in gasolene, without damaging them.

43 Meters at present under construction are being made with the axis of the cylinder vertical, with a view to greater convenience of access and in making connections.

44 The first large meter of this type to be installed was put in the works of the Milwaukee Gas Light Company, and the writer is indebted to the officials of that company for their coöperation in making extensive tests during the work of development.

45 Referring to Par. 16, for gas or air under the conditions existing during the tests, of approximately 60 deg. fahr., 29.8 in. mercury and 6 in. water pressure, the correction for water vapor introduces a change in the results of less than one-half of one per cent, and was therefore omitted. At other pressures and temperatures the correction for water vapor can be easily made by reference to the charts commonly used in gas works. An interesting confirmation of the statement in Par. 16 appeared during the tests, in that the most minute addition of electrical energy caused an immediate rise of temperature of the gas or air. This was repeatedly tried with great care, and always with the same result.

THE TRAINING OF MEN—A NECESSARY PART OF A MODERN FACTORY SYSTEM

By Magnus W. Alexander, Lynn, Mass.
Member of the Society.

Emerging from the depression of the last two years, American industries are once more entering upon an era of prosperity, which in the natural course of events should surpass in magnitude and intensity anything yet seen in the industrial world. One obstruction alone lies in the path of this unrivaled future: lack of men to do the work is the fact that confronts keen observers of the situation. There is no lack of enterprise in the country; money for sound business undertakings is plentiful; and the consuming capacity at home and abroad is increasing from year to year. But are we in a position to utilize these factors to the fullest extent?

2 Only a few years ago the cry for efficient men in all branches of industrial activity was universal and insistent, and manufacturers everywhere complained of their inability to man their establishments properly. Skilled mechanics were at a premium; capable industrial foremen and superintendents were painfully scarce; while positions of leadership calling for men of education, experience, and breadth of view could be filled only with difficulty.

3 The industrial depression of 1907–1908 naturally relieved the embarrassment; but even then skilled mechanics and efficient foremen could not be secured in adequate numbers. The last few months have already clearly demonstrated that the acuteness of the situation has returned, and that this condition will be accentuated as time goes on. Should we, then, not profit by the lessons of the past and cast about for an adequate remedy? Now is the time to analyze the situation, and, in the light of our experience, work out a comprehensive policy which will enable us to cope with the exigencies as they arise. This is a matter which concerns every manufacturer, large and small; it is as much a problem of business sagacity as of immediate necessity.

4 In December 1906, I had the privilege of presenting to The American Society of Mechanical Engineers my ideas concerning the train-

ing of young men for positions as skilled mechanics and foremen, and of showing how this scheme had been put into practical operation through the apprenticeship system of the General Electric Company at West Lynn, Mass. In the meantime this system has been materially extended so as to provide adequately for the boy with a grammar school, a high school, or an engineering college education. New lines of factory work have been included, and cognizance has been taken of the necessity for training machine specialists. The educational scheme of the Lynn Works, therefore, presents in its present scope a comprehensive policy.

5 The underlying thought of all this training is the belief that skill will demonstrate its full potential value only as it is supported by intelligence. Each course of training, except in the case of machine specialists, includes, therefore, distinctive educational work, and the scope of each course is based on the previous education of the individual. There are, of course, young men of pronounced native ability, who, no doubt, would prove to be efficient in training courses from which the above educational requirement excludes them, but to deal with these exceptions would complicate the process of selection and the system of training. In the training courses for trade apprentices alone, which are ordinarily open to grammar school graduates only, boys with an incomplete grammar school education, who can pass a satisfactory examination, may be admitted; in all other courses rigidity of requirements is necessarily maintained.

6 In the training leading to positions as machine specialists, such as shaper, lathe and boring mill hands, milling machine operatives, etc., no provision has yet been made for educational advancement as distinctive from training for skill. This can be effected at any time, however, without undue expenditure.

7 The company recognizes the existence of workingmen who are in the class of unskilled labor from lack of opportunity or of foresight or due to other circumstances not under their control and for the same general reasons remain in such service. Many, of course, by disposition and general makeup, are bound to find their livelihood in such unskilled labor, while on the other hand, many can be trained in a comparatively short time to semi-skilled and skilled special work. Such training will increase their economic value and their contentment and add materially to the productive efficiency of the factory.

8 In pursuance of this policy a systematic effort is made to select from among the unskilled workers men of from 20 to 35 years of age, who give fair promise of success as machine specialists. Some of

these men are now receiving instruction in lathe work, others in shaper or boring mill, or planer or milling machine work. This training lasts from three to four months, depending on individual capacity, and the men receive during that time an hourly rate which gives them a living wage. A capable instructor makes the selection, assigns the men to the various factory departments where machines and work are available, and supervises their training.

9 These men are, of course, under the foremen in whose department they are working for the time being, but the instructor, who is a very capable skilled mechanic, having had charge of men for many years, visits them almost daily and sees that they receive work of an instructive character and of advancing difficulty, as far as this can be done without undue interference with the productive requirements of the factory. When the instructor is satisfied with a man's capability of handling his machine and of turning out a fair amount of work, he assigns him permanently to a foreman who requires such service. The machine specialist then takes his place as a regular workman and receives regular day or piece work compensation. The same man, however, may apply again to the instructor for special training on some other machine; thus gradually fitting himself for a position as all-around machinist and tool-maker, with correspondingly higher compensation.

10 An arrangement of this kind entails no material hardship on anyone, gives many men an opportunity to rise to a higher plane of efficiency, automatically supplies the factory with capable machine specialists, and tends to attract to the factory men of ambition and stamina. This work might be further extended by giving to those who cannot sacrifice the temporary reduction of wages, an opportunity to receive their training during evening hours and on Saturday afternoons. This problem was outlined in my paper, A Plan to Provide for a Supply of Skilled Workmen (Transactions, vol. 28, p. 439).

TRAINING OF MECHANICS, FOREMEN, DESIGNERS, ETC.

11 Far more comprehensive in scope, and covering a longer period of time, must of course be the training of those who are to take positions as highly skilled mechanics and foremen, designers and engineers, superintendents and managers. The industries themselves must furnish this training, inasmuch as our school systems do not provide for it today, and very likely in the future will be able only to approach the full requirements. Mental training closely correlated with prac-

tical instruction may be gained by putting both under the sole charge of the factory management; or smaller factories may combine for joint classroom instruction; or the theoretical instruction may be delegated entirely to public school authorities, who could provide special classes for instruction alternate days or weeks. All three schemes are in operation today, and either will prove effective if properly managed, and if selected with reference to local conditions, size of factory and available personnel.

12 A bare outline of the system established by the General Electric Company, at West Lynn, which was fully treated in my former paper, may be of interest as showing how it has developed in the last three years, during which time about one hundred apprentices have graduated from the course.

REGULAR APPRENTICE TRAINING

13 Boys of at least 15 years of age, who have had a grammar school education or its equivalent, may be admitted on completion of a two months' trial period, to the regular apprentice course. It is largely contended by manufacturers that boys under 16 are not fit for trade training. A normally bright boy, however, unless he goes to high school, will usually be obliged to seek employment at 15, and it is better for him to be put immediately under systematic trade instruction. Naturally, the work at the beginning must be suited to the boy's immature physical as well as mental development, and boys lacking in physical strength will be accepted neither at 15 nor at 16 years of age.

14 The training for future tool and die makers, instrument makers and pattern makers, lasts four years, while iron, steel and brass molders, blacksmith and steam-fitter apprentices, who should be somewhat older and stronger, receive three years of training. The two months of trial are included in this period. Apprentices receive compensation, even during the trial period, at the rate of 8 cents per hour for the first six months, 10 cents for the second six months, 12 cents for the second year, 14 cents for the third year, and 16½ cents for the fourth year. Molder, blacksmith and steam-fitter apprentices, on the other hand, receive 10 and 12 cents per hour respectively for the first and second six months' periods, 14 cents for the second year, and 16½ cents for the third year. In either case satisfactory completion of the course entitles the graduate to a Certificate of Apprenticeship and a cash bonus of $100. The normal number of working hours is 55 per week.

15 The average compensation paid to graduated apprentices is $2.75 per day, although some are started at $3 a day immediately upon graduation. The significance of these figures is more fully appreciated when it is borne in mind that the young man of 21 years receiving such pay is only just beginning his life's work, with a solid preparation for marked future advancement.

16 All apprentices are obliged to spend from an hour and a half to two hours in the classrooms every day except Saturday, except during part of July and August, when instructors and apprentices may take their vacations. Classes meet during regular working hours, usually at the beginning or end of the half-day periods. Full compensation is paid during classroom hours. Retention on the course and the payment of the bonus are dependent on satisfactory work in the classroom as well as in the shop, and the standing in both is stated on the Certificate of Apprenticeship.

17 The classroom instruction is based on a grammar school education, and includes arithmetic, algebra, geometry and trigonometry, physics as it concerns simple machines, power transmission, strength of materials, machine design, magnetism and electricity, mechanical drawing, and jig and fixture design. For pattern-maker and molder apprentices an extended course in mechanical drawing is substituted for tool design. This instruction is practical, with constant reference to the work of the apprentices and to the usual factory problems, the aim being, above all else, to develop the ability to reason, and to foster a pride of vocation.

18 In no way is this stimulated more than by the daily practical talks of the superintendent of apprentices, who carries, so to speak, the factory into the classroom. The many answers offered by the apprentices to such a question as, "Why does a one-inch drill cut a larger hole in cast iron than in steel?" reveal their mental capacity and mechanical understanding and give the superintendent a splendid opportunity for driving home practical truths. The superintendent continues this kind of instruction in the apprentice training room. If he notices, for instance, that an apprentice uses an improperly ground tool, he calls a number of the boys to the blackboard and explains clearly by means of sketches what is wrong about the tool and how it should be sharpened.

THE APPRENTICE TRAINING ROOM

19 The training room is a special department for apprentices, a trade school in the factory, with this distinction, however, that all

work is commercial work selected solely for its instructive character. It had its inception in the belief that the apprentice should receive his initial training under the most favorable conditions and expert supervision. The very fact of this work being a part of the commercial output of the factory automatically insures a high standard of quality and quantity, and eliminates the false notions of these values usually found in purely educational trade schools. As a matter of record, the work of the apprentice is of a very high standard. Moreover, on work of a repetition character, the apprentices attain a speed of from two-thirds to three-quarters of that of the average workman, and a quality of work fully equal to the average; while on work generally classed as tool work, the apprentices very closely approach and sometimes even equal the work of the skilled journeyman.

20 The reports of the general inspection department show that rejected motor shafts, for instance, average only 2 per cent. although the permissible limits for the journal and other parts of the shaft are usually not more than 0.0005 in., and in any event not more than 0.001 in. Other work requiring accuracy to micrometer measurements is equally creditable to the apprentice training. Wherever possible, jigs made by the apprentices are not allowed to leave the training room unless the accuracy of the work has been proved by drilling or machining a part for which the jig was made. Several molds for various materials have been recently finished in the apprentice training room and the accuracy of the work proved not only by the pieces molded, but also by the fact that the parts of the various molds could be accurately assembled in the different permissible combinations.

21 Training rooms have been established for tool-maker and pattern-maker apprentices, occupying departments of about 15,000 sq. ft., and 4000 sq. ft. respectively. No training room has yet been organized for molder apprentices, of whom there are only a few, this part of the system being not yet very far developed. The training rooms are in charge of expert mechanics who act as assistant foremen to the superintendent of apprentices. One assistant takes care of about 25 apprentices in the pattern training room, and four assistants look after the business conduct of the machinist training room, with the instruction of about 130 apprentices. The small number of instructors and supervisors is explained by the arrangement under which the apprentices themselves, at various stages, act as instructors to those less advanced. In this way, not only is the instruction carried on with economy, but latent ability for executive

work is developed and the apprentices are taught self-reliance much more quickly than if their every step was directed by journeyman instructors; the aim being to train skilled and intelligent mechanics, as well as to develop on this basis industrial foremen.

22 It is indicative of the individual instruction afforded, that not infrequently a boy teacher has served a shorter period of apprenticeship than the pupil he instructs. No course has been laid out for the practical work; each apprentice being advanced as fast as is consistent with his individual capacity. He must have a fair understanding of his machine and be able to produce his work with commercial accuracy and a fair degree of speed before he can be advanced.

23 The company believes that inasmuch as it pays good apprentice wages and offers excellent training and educational advancement, it is justified in expecting a high standard of workmanship and of deportment. Accordingly, a rigid weeding-out process takes place throughout the course; more than 50 per cent of those serving the trial period are dropped at the end of two months and quite a few are discharged even after having signed the apprentice agreement. At first, a provision was made to send the apprentices to different departments in the factory, after about a year in the training room; later on, the time in the training room was extended to two years and the tendency now is to increase it to about three years before giving the apprentice a change to acquire additional experience. The advantages of this arrangement lie not only in the extended systematic training of the apprentices, but also in their better general supervision during the most impressionable period of their lives. At times, of course, apprentices in all stages of training are loaned to factory departments for a few days or weeks; on the other hand, some of those who have already progressed into the factory are brought back into the training room, if the quantity or quality of their work, or their deportment, necessitates such disciplinary measures.

24 At the present time there are over 200 trade apprentices at the Lynn Works, while 101 have already graduated. Of these 63 are now in the employ of the company, 8 serving as assistant foremen, 5 as inspectors and 12 as tool draftsmen, while the remainder work as skilled journeymen. Many of the latter, no doubt, will rise to positions of added responsibility during the next few years. The point is made clear to all apprentices, however, that a position as foreman or superintendent should not be the sole aim except for those with predominant executive ability. The percentage of graduates remaining with the company—in many respects a measure of the success of

the scheme—varies, but it has never dropped below 55 and at times has been over 80.

DRAFTSMAN APPRENTICES

25 No less encouraging are the results achieved with draftsman apprentices. Training for positions as draftsmen and designers is limited to young men with a complete high school education who pass examinations in algebra, plane geometry and elementary physics. It has been found necessary to introduce this examination on account of the great divergence in the curricula of high schools and the great difference in scholarships among graduates. Accepted applicants must serve a two months' trial period satisfactorily before being indentured for an apprenticeship of three years at 10 and 12 cents per hour respectively for the first and second six-month periods, fifteen cents per hour for the second year, and twenty cents for the third year. They receive then a cash bonus of $75 and a Certificate of Apprenticeship which states their efficiency in practical and in theoretical work.

26 Draftsman apprentices are obliged to attend classroom exercises about an hour and a half every day, except on Saturday and during part of July and August; and a considerable amount of home study is required. The educational work consists of advanced algebra, descriptive and analytic geometry, plane trigonometry, advanced physics, inorganic chemistry, strength of materials and machine design. Instruction is for the most part of college rank, and college text books are used entirely, but again the closest correlation with the practical work in the shop and drawing office is maintained.

27 Examinations are held three times a year and failure to pass in all subjects with at least 70 per cent necessitates the repetition of a fourteen weeks' period. A second failure in the same subjects, or repeated failures in different grades, result in the discontinuance of the whole course. This does not necessarily mean that the delinquents must leave the company, for under certain conditions they are permitted to continue on the shop apprentice course under a four years' agreement. A few have already made this adjustment.

28 Draftsman apprentices receive machine shop training during the first year and a half, and drafting instruction during the remainder of the three years' course. The machine shop work is given principally in the apprentice training room on account of special facilities for this instruction; a part of the time, however, is devoted to repair

work on machinery, and to tool work. The object is to give the future draftsman and designer an adequate insight into practical work so that he may appreciate in his designs the possibilities and limitations of the shop, and may, moreover, bear in mind the use of jigs and fixtures for economic manufacture on a large scale. The shop work, finally, inculcates in the young man an appreciation of the value of time and money such as he would not easily acquire without this training.

29 The work in the drawing office begins with a brief period of tracing, for the purpose of teaching the use of instruments, neatness, and the general arrangement of shop drawings; it continues on detail drafting and finishes with layout and design work. The high quality of the work of the apprentice is the natural consequence of the careful selection of applicants and the enforcement of a high standard of practical and educational achievement. Most of the young men are indeed a credit to the system.

30 The course for draftsman apprentices was originated about six years ago, but on a less ambitious plane. Grammar school graduates were then admitted for a four years' training, and no particular stress was laid on educational instruction. It was soon found, however, that in this way good tracers and detail draftsmen could be developed, but not high-grade draftsmen and designers. About four years ago a high school education was made an entrance condition, and a course of three years was offered. Soon after, class room instruction was added, extending but slightly beyond a review of the high school program. The apprentices who had been admitted under the less exacting requirements naturally fell behind and had to drop out, and the standard of the educational and drafting work was then gradually raised. Thirty-two apprentices have been graduated under these conditions, most of whom have become competent draftsmen, while a few have started on promising careers as designers.

31 Still the company was not satisfied with the scope of the course. It was recognized that the general standing of a draftsman and the standard of his work had everywhere deteriorated during the last decade, largely on account of the great influx of superficially prepared draftsmen. To develop draftsmen and designers of pronounced capacity and intelligence would dignify the work and regain full recognition of its potential importance. Moreover, the graduates of the machinist apprentice course proved to be capable of developing into competent tool and mechanial draftsmen, so that the demand for high-grade, intelligent designers became the more

pertinent. The final change in the course was therefore made, about a year ago, calling for an entrance examination, for educational work of collegiate grade, and for the extended training in shop work.

32 Ten draftsman apprentices are finishing their apprenticeship under the less exacting conditions, while twenty-four are receiving their training in the new course, and their progress augurs well for their future. In fact the training under the prevailing rigid system is expected to prove so effective that the privilege of a one year's post-graduate course in the various testing departments will be extended to all draftsman apprentices whose shop and office work has been very satisfactory and who have a standing of at least 85 per cent in all theoretical studies. These young men will then be eligible for important positions in engineering or commercial organizations.

33 Two other opportunities for systematic training have recently been opened to high school graduates, one for the preparation of testers and erectors of machinery, the other leading to a business career in a manufacturing establishment. These courses were instituted only within the last few months, and the results can only be foreshadowed.

TESTER AND ERECTOR APPRENTICES

34 Tester apprentices must pass an examination the same as draftsman apprentices, and the length of the two courses and the rates of compensation are identical. The practical work consists of about six months of testing motors or transformers, followed by about nine months of assembling and winding, the remaining year and nine months being devoted to a training in the various testing departments for meters and instruments, arc lamps, rectifiers, railway motors, and special electrical machinery, also turbines and turbo-generators. This work will be carefully supervised by a competent instructor, who will arrange for transfers from one class of work to another, and who will constantly keep the tester apprentices up to the required standard.

BUSINESS APPRENTICES

35 Business apprentices are recruited from high school graduates who have a leaning towards business activity, but not sufficiently high scholarship to pass the entrance examination and continue the educational work prescribed for the drafting and testing courses. These apprentices enter upon a two years' course at a compensation of

12 cents per hour for the first year and 15 cents for the second year, with a cash bonus of $50 at the successful termination of the course. They begin with six months of general stockkeeping, which acquaints them with the principal materials used in the factory, and leads to an appreciation of the value of these materials. Then follows a nine months' training on the writing of material lists and the compiling of stock reports. In this way the apprentices learn to read drawings and to make up lists of the kinds and amount of materials required for the production of one or several machines or machine parts delineated on drawings. They learn, furthermore. to calculate the losses that result from the cutting off of bars and the punching of various shaped parts. All this leads to accuracy and develops an interest in the value of stock which will show in the stockroom work of a more independent character which occupies most of the remaining nine months. These apprentices will also be given a training in shop clerical work, and this will be supplemented by instruction in arithmetic and geometry, the reading of drawings, and simple bookkeeping. This course has been instituted because the great value of proper stock keeping and factoıy accounting is recognized.

STUDENT COURSES

36 So far the plan outlined has dealt with methods of increasing the industrial efficiency of workmen and of preparing boys with a grammar or high school education for the trades and for semi-professional service. The company also seeks to provide, for young men of engineering collegiate training, an entrance into the industrial field which will lead to positions of scientific importance and administrative responsibility.

37 In common with other manufacturers the General Electric Company established many years ago, a "student course" providing for a two years' experience in the various testing departments of the Works. The young men were usually assigned to a testing department for a certain time, and were then more or less automatically transferred from one department to another until they had covered the whole course within the specified time. They very often did not take the work seriously enough, and in any event their chief aim was to get a general knowledge of as large a field as possible, rather than to acquire thoroughness in each specific field.

38 The company endeavored to eliminate these defects by organizing some three years ago a supervisory committee. This committee

met frequently with the students, and examined each one once or twice a year, in order to test his theoretical and applied technical knowledge, and his alertness for taking full advantage of the educational opportunities offered. In this way the committee was enabled to weed out some who showed no capacity for future responsible work, and to modify the course in each case to fit the individual student, and lead him into the field of his greatest probable usefulness. This arrangement unquestionably improved the general standing of the student training.

39 The real value of the committee's work, however, has been in the close contact of the members of the committee with several hundred students, and the opportunity to study carefully and specifically the bearing of the student training upon the work of the graduates in various positions inside and outside the factory organization. The committee soon recognized that while the training above described was, in general, a good preparation for those who elect positions in the selling organization, it did not give the right kind of experience to those who are to become designing, manufacturing and administrative engineers. This latter group needs a far more comprehensive knowledge of mechanical processes and a better understanding of the economic forces at work in a modern industrial establishment than can be acquired in the testing departments. With this in mind, the student course was put on a new basis a year and a half ago, and the work so far accomplished bears out the correctness of the premises.

40 In the present form the student course is divided into two parts, the so-called engineering and the commercial course. Admittance to either is dependent on a complete engineering college education, and almost invariably applicants must appear personally before the committee. The courses last two years and the compensation has been set at 20 cents per hour ($11.00 per week) for the first year, 22½ cents per hour ($12.37) and 25 cents per hour ($13.75) respectively for the two halves of the second year. Without written agreement it is mutually understood that the student will give to the company two years of faithful service, and that the company, on the other hand, reserves the right to terminate the work of any student who at any time proves that he is not above the average either in capacity and special fitness or in good intentions. Aside from the direct advantage of such a rigid arrangement, is the added result of attracting to the course, as has already been demonstrated, high-grade young men, some even with one or two years of practical experience after graduation from college, who aspire to positions of prominence and realize

the value of a stiff training course with correspondingly good prospects. Even in busy times, when college graduates are in demand everywhere, young men with inherent capabilities will gravitate toward the Lynn course or any other course of equally high order.

41 Weekly evening lectures have been arranged, which all students are expected to attend, when the engineers and foremen of the company, as well as heads of the business departments, informally address the young men and stimulate a free and frank discussion of the subject under consideration. These lectures cover the principal materials of construction, important manufacturing processes, and the various lines of apparatus manufactured by the company. Occasionally talks dealing with business methods are interspersed. The complete program includes lectures on: (1) Iron foundry practice; (2) Steel foundry practice; (3) Pattern making; (4) Alloys and their properties; (5) Stockroom methods; (6) Forging; (7) Hardening; (8) Welding; (9) Factory cost keeping; (10) Tool steels; (11) Shapes of cutting tools; (12) Cutting speeds and feeds; (13) Piece-work rating; (14) Fibrous insulating materials; (15) Oils and varnishes; (16) Porcelain and molded compounds; (17) Shop bookkeeping; (18) Wires and cables; (19) Selection of materials in reference to design; (20) Drawing office methods; (21) Principal machine tools; (22) Care of and repairs to machinery; (23) Interchangeability of parts; (24) The essentials of production; (25) Punch press operation; (26) Die making; (27) Automatic machine processes; (28) Distribution of labor charges; (29) Requirements of accuracy in machine work; (30) Arc lamps; (31) Incandescent lamps; (32) Mercury lamps and rectifiers; (33) Lighting systems; (34) Factory building construction; (35) D. C. and A. C. motors; (36) Electrical features of motors; (37) Mechanical features of motors; (38) Motor drive of machine tools; (39) Labor report and pay roll; (40) Fan motors; (41) Industrial motor applications; (42) Railway motors; (43) Gears and pinions; (44) Avoidable factory losses; (45) Meters and instruments; (46) Standardization of instruments; (47) Sheet iron for electrical machinery; (48) Annealing of iron; (49) Transformers; (50) Testing of electrical machinery; (51) Shipping and receiving methods; (52) Steam turbines; (53) Valve gears and governors; (54) Buckets and bucket wheels; (55) Turbo Generators; (56) Turbine testing; (57) Wage payments; (58) Centrifugal compressors; (59) Gas motors; (60) The labor problem; (61) The reading of technical magazines; (62) Salesmanship; (63) Factory management.

42 Lectures are illustrated, by samples of materials and machines,

pictures, drawings and charts. They put the students in possession of up-to-date, practical information which they are not able to get from books or from outside sources. Lecturers and students alike have expressed their enjoyment of these evenings, which inculcate a certain class spirit which results in added ambition, increased loyalty to the employer and a broader conception of the work. This class spirit is discouraged, on the other hand, during the daily work, that the students may never forget that they must earn, as well as learn, in the service of their employer.

COMMERCIAL STUDENT COURSE

43 Commercial students spend about 2 months on meter and instrument testing, 2½ months on arc lamp testing and repairing, 1½ months on transformer winding and assembling, 4 months on transformer and rectifier testing, 2½ months on stationary and railway motor winding and assembling, 6 months on stationary and railway motor testing, and 5½ months on turbine testing or on other special assignments. They are stimulated to keep in touch with the latest engineering developments by carefully reading the technical magazines, and are shown the value of following up the advertisements in them as one means of getting acquainted with the general features of apparatus manufactured by competitors. Finally, the students are assisted in visiting power stations and installations, where they may see apparatus of various manufacturers and learn to observe keenly the essential points of operation.

44 Engineering students, on the other hand, receive most of their training in the machine shops, winding departments and drawing office, while the latter part of the course is devoted to testing, or to production, cost or other business activities as the capacity and inclination of the individual student may make advisable. These students are usually started on machine work in the apprentice training room, where they can receive instruction under the most favorable conditions for the first month or two. During this time, also, they can be closely watched, and here the first process of elimination takes place. In the next eight or nine months, students are assigned to various departments of the factory, in some of which they are put entirely on production work, in order that they may come under the influence of the intensity of production and may learn the possibilities of output on various machines, and in others they get experience in tool-making and repairs to machinery. Students who show an inclination toward heavy work are usually assigned to departments

in which the machining of turbines, street-car motors, and large motors in general, is done. Students inclined toward light work are, on the other hand, transferred to machine and tool-making departments in the arc lamp, meter and instrument or fan motor building. Some of the students spend a month or two in winding and insulating departments, again with particular reference to their future specialty.

45 For the following ten months, approximately, students are assigned to the drawing office, where they work first on detail drawings and then on assembly and layout work. Part of this time is devoted to tool designing, when students learn to design a drill jig or milling fixture or similar auxiliary apparatus for economic wholesale manufacture. The advantages of the drafting experience are obvious, especially for those who wish to become designing and manufacturing engineers. Inability to read drawings quickly, with an eye that sees the delineations grow into shape and form an achievement which can usually be gained only through a somewhat extended drafting experience, prevents many college-bred junior engineers from occupying positions of responsibiltiy in designing and manufacturing work. Drawing is the language of the engineer; it is equally useful to the one who supervises draftsmen and designers, or who interprets shop drawings to the mechanic and the foreman, and to the one who wishes to sell a piece of apparatus, when, by means of sketches he can illustrate the advantageous points of manufacture.

46 The remaining four or six months of the course are devoted to specific work leading to some definite occupation after graduation from the course. Thus, if the committee and the student agree that his future work should lie along manufacturing lines, he may act for a month or two as assistant to a department foreman, and acquire additional specialized shop experience. Another student, better fitted for scientific research or for general mathematical work, may receive a few months' experience on testing, especially of experimental apparatus, and may temporarily be assigned to an engineering department. A student who has shown particular aptitude for commercial work may be given some production and cost-accounting experience, while the future salesman is given an opportunity to spend the remainder of his course in various testing departments. A strong point is made of studying the development of each student, week by week, in order to train him along lines of his greatest capacity.

47 This purpose, as well as the desire to assist every student in the best way possible, and at the same time to exact from him a full

measure of service, has led to the appointment of a special instructor whose function it is to keep in almost daily touch with every student throughout the plant. The instructor endeavors to make the foremen of the departments, to whom students are assigned, sympathetic with the whole educational scheme, and to secure for the student work that will be especially helpful to him; he sees to it that every student works at the highest point of efficiency, and whenever he finds him doing his work in anything but the most approved fashion, or using wrongly sharpened tools, or fine feeds where coarse feeds are the proper thing, he explains and insists on remedial action. He, furthermore, tries to inculcate in the student a proper conception of his work, and to make him feel that while he must at all times give service, some one who is sympathetic stands ready to assist him.

48 The instructor makes a written report of the work of each student once a week, and presents it at the weekly meeting of the committee. The committee discusses every student in the light of the report, lays out his course for four weeks or four months ahead, as the case may permit, and talks personally to those of whom the instructor is not able to report favorably. An admonition is usually considered equivalent to placing the student on a few weeks' probation, with the understanding that he will be dropped without hesitation if at the end of the probation period a decided improvement cannot be reported. The committee, furthermore, interviews new applicants, and selects those for the engineering or commercial course.

49 A competent instructor with testing experience will soon be appointed to follow the commercial students through their course, unless the commercial course is abandoned on the theory that students with a training such as the engineering course offers will be better salesmen than if they had testing experience alone.

50 All in all, it would seem that the student training has been laid out on a broad basis, with due regard to the interests of the student as well as those of the company; and it is fair to expect that this training will develop a body of theoretically and practically educacated young men, who, on account of their knowledge, their broad conception of things, and their sympathetic outlook, are in line for positions of the highest order either with the company or with other concerns.

51 A definite policy, a sympathetic following up of the students, insistence upon a high standard of work, and a sympathetic oversight of the students by a committee of competent men, are the distinguishing features of the Lynn student courses.

52 The educational policy of the Lynn Works provides systematic training suitable to all classes of people. The unskilled worker without particular education receives a training adequate to his immediate needs; the grammar school boy is initiated into the trades on the basis of a four years' course with educational instruction of a high school character; the high school graduate is trained for semi-professional service of a technical or business nature, on the basis of a three years' course with educational instruction of collegiate grade; and the college graduate is prepared for professional service of the highest order, on the basis of a two years' training of which the educational instruction assumes the character of a post-graduate college course. Obviously, there are other ways of obtaining these results. Coöperative efforts between the engineering college and the factory, for instance, may be substituted for college instruction, followed hy practical training through a student course. (See address before the American Institute of Electrical Engineers, June 1908, A Method of Training Engineers.) Educationally, psychologically and economically, the scheme is sound.

53 There are three main problems that enter into production,—the machine problem, the material problem, and the man problem. It is clear that the man problem is the most difficult of solution but also the most important in competitive activity. It must be approached in the same scientific manner and with the same painstaking concentration of effort that is today applied to the other two problems. The training of men must be the key-note of our industrial expansion. At least in the larger industrial establishments, this calls for a new type of engineer who might appropriately be known as the economic engineer.

DISCUSSION

THE HIGH-PRESSURE FIRE-SERVICE PUMPS OF MANHATTAN BOROUGH, CITY OF NEW YORK

By Prof. R. C. Carpenter, Published in the Journal for September

ABSTRACT OF PAPER

This paper describes the high-pressure pumping systems installed for fire service in the city of New York and gives the results of tests of the pumping machinery. There are two pumping stations for the system located in different parts of the city, deriving their supply from the Croton system, although sea water can be used in an emergency. There are five pumping units in each station consisting of Allis-Chalmers five-stage centrifugal pumps driven by induction motors. The pumps each have a capacity of 3000 gal. per min. and a delivery pressure of 300 lb. per sq. in. The distribution system covers a large section of the city between Chambers and Twenty-third Streets and is designed for the high pressure that must be met in service. In the tests the quantity of water discharged was measured by venturi meter. Tests were made of the pumps when running together and of certain of the pumps running singly under different discharge pressures. The efficiency of the pump tested singly in one of the stations under normal conditions varied from 70 to 77 per cent, and of the pump tested in the other station, under the same conditions, from 76 to 79 per cent. The pumps were put to a crucial test on January 7, 8 and 9, 1909, when brought into service for five simultaneous fires. Seven pumps were operated, delivering 35,500 gal. per min. against an average pressure of 225 lb. at the pumps and 205 lb. at the hydrants. The total pumpage was 14,095,000 gal., and the current used 81,450 kw-hr., costing $1222.

DISCUSSION AT NEW YORK

Prof. George F. Sever.[1] The electrical features of this installation are of much interest but the reasons for selecting that system which is now in operation should be given. In the discussion of this problem both alternating and direct-current power were considered for the operation of the motor-driven pumps, and alternating-current power was decided upon. The reasons for such selection I have noted herewith:

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a Absolute simplicity, that being the key-note of the electrical end of this power installation.

b The absence of all commutating apparatus and brushes.

c Induction motors provide very quick starting when it is necessary to operate the station on a fire signal.

d There is less expense for copper in the distribution system to insure continuity of service.

e The induction motor is a less expensive apparatus than the direct-current motor.

f With the induction motor there are absolutely no exposed live circuits in the station, as there might be with a direct-current apparatus. The final decision was for 3-phase service at 6600 volts and 25 cycles. It was decided that it would not be desirable to establish a power house to be operated by the city because it would be a municipal plant.

2 In order to insure continuity of service there is brought to each pumping station an independent feeder from each of the two Waterside stations of the New York Edison Company. There is also brought to each pumping station an independent feeder from the nearest substation of the New York Edison Company, as follows: to the Gansevoort Street station two feeders from the Horatio Street substation, and to the Oliver Street station two feeders from the Duane Street station of the company. Hence there are really four independent sources of power supply for each pumping station, assuring practically no possibility of shutdown.

3 The contract for electric power for the Manhattan station was let to the New York Edison Company. This contract provides for two payments, the first for a reservation of 3250 kw. capacity, of generating, distributing and controlling apparatus, available at either pumping station at an instant's notice, or practically without any notice at all. Thus four pumps can be thrown on with absolutely no notice to the New York Edison Company that they are to be used. For that reservation, and care and maintenance of the whole distributing system, the city pays about $63,000 per year, and the city also pays one and one-half cents per kw-hr. for all high-tension power used in each station.

4 There is also another interesting stipulation in the contract, which may be of interest to the engineers as it provides for the protection of the city. This stipulation is as follows: "If the contractor, under the terms of this contract, shall fail to maintain and deliver

a continuous and uninterrupted supply of electric power when required, the contractors shall and will pay to the city the sum of five hundred dollars per minute for each minute's interruption or delay of electric power supply after the power has been interrupted or delayed for three consecutive minutes." So, if they cannot deliver power after an interruption of three minutes, immediately a charge of $500 per min. is imposed and is deducted from the bills which the New York Edison Company renders.

5 The operation of both these stations is extremely simple. The handle of the oil switch is turned, throwing the 6600 volts directly on the stator of the motor. By turning a hand wheel, the motor is brought up to speed in less than 33 sec., and in starting the current is not supposed to exceed 150 per cent of the full-load current, which is 64 amperes. As far as I have observed the operation of the station, there has been absolutely no trouble from the electrical end, no trouble with the feeder system, and none with the motors, and I think the City of New York has two plants which will give them for many years to come absolutely no trouble whatsoever.

Wm. M. White. The paper deals with questions in which I am directly interested. The methods employed in making the tests were probably the best that could have been selected. There is probably no more accurate method of determining the quantity of water delivered by a pump than by the venturi meter, especially when in the hands of an expert who is familiar with its workings. The venturi meter, as Professor Carpenter says, has been used for a number of years; it has been tested in various ways and proved to give accurate results. The power delivered to the pumps can be most carefully obtained by electrical instruments.

2 The writer accepts without question the various efficiencies obtained and presented by the author, who states, calling attention to the variation in efficiencies obtained, that the individual observations do not agree as closely as he would like. I do not think Professor Carpenter should offer any apology as the results seem to agree very closely, and certainly are as accurate as are generally obtained on work of this kind. The efficiencies obtained on these pumps, though not the highest that have been obtained, are as high as is usual for similar conditions of head, capacity and speed. The designers of the pumps deserve credit for the performance shown by the pumps.

3 I am at a loss to find a reason for the variation in efficiencies of the pumps, as mentioned in Par. 65, where it is stated that individual

pumps delivered 10,000 gal. per min. at a velocity of 1100 ft. The efficiency was thus between 65 and 70 per cent, although in later tests made by the Government, when nothing but water passed through the pipes, the efficiency rose to as high as 80 per cent.

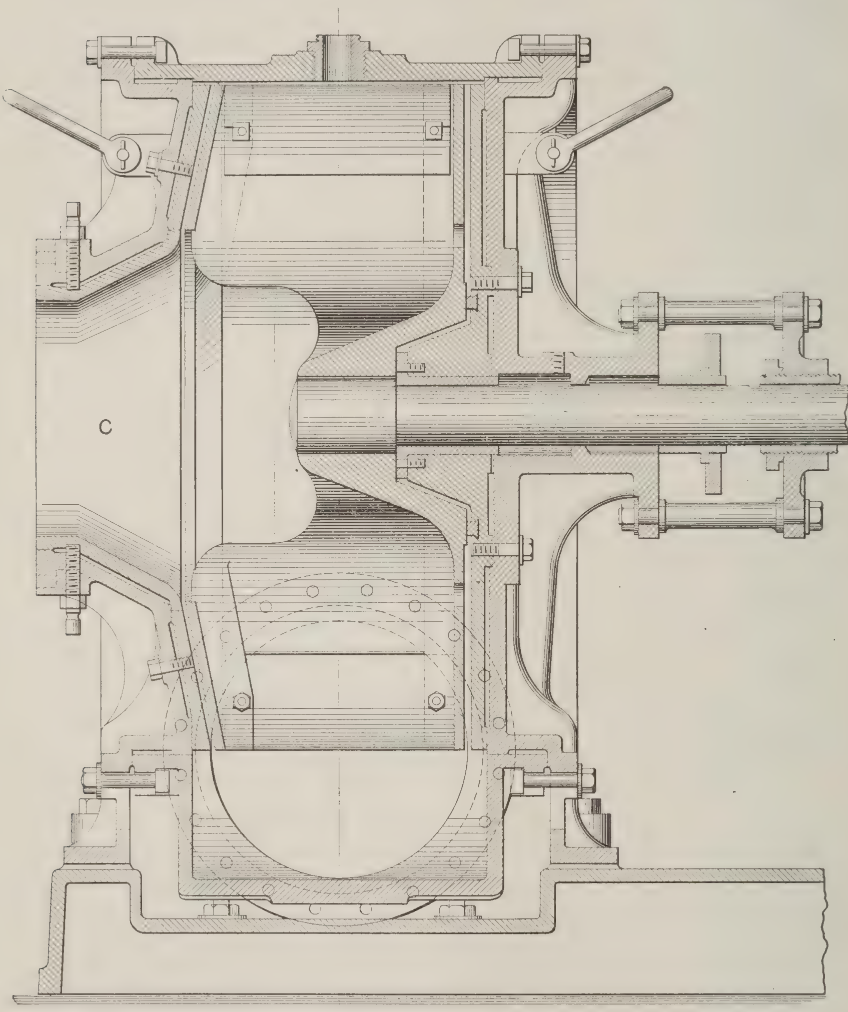

Fig. 2 Sectional View of Centrifugal Pump for Dredging

5 The shoe used is a hook that drags along the bottom, chains being fastened to the vessel for this purpose. The vessel never

stopped from morning to night, simply running out to sea, dumping, and coming back again to work.

6 At the point L, Fig. 3, was the heavy shoe that served to dig into the mud and gravel. At O was a butterfly valve, kept open all the time to admit water above the drag to mix with the material raised. At the bottom K was another valve which could be opened in an emergency, in case not enough water was admitted at O.

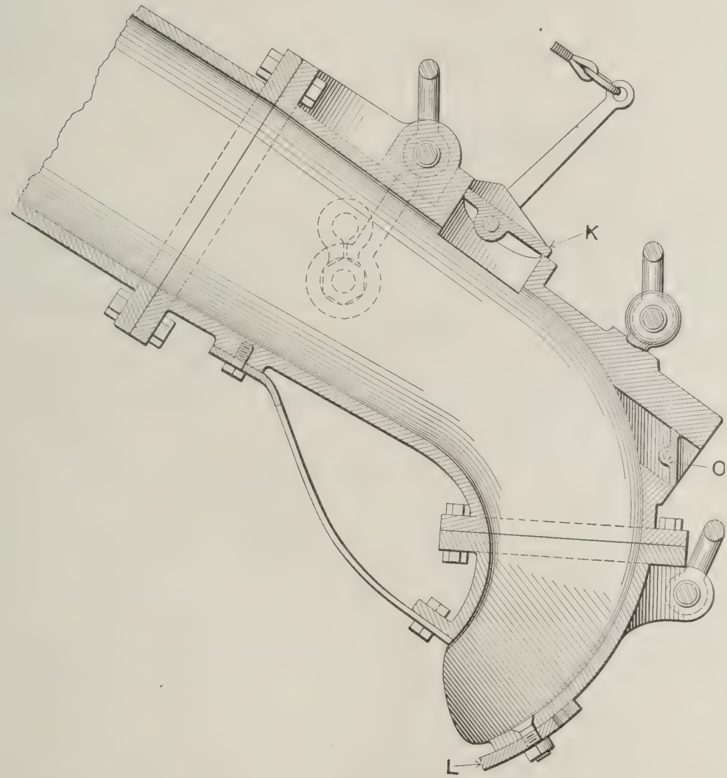

Fig. 3 Detail of End of Suction Line

7 The pump itself was of a plain centrifugal type, 40 in. in diameter, with vanes cut away at the center, as shown in Fig. 3. Because of this arrangement, the material would come in at C and out of the vanes at the discharge, without damaging the pump when heavy substances were drawn in. The three vanes were made with wings

bolted on, and accessible from both sides. The thrust was taken up by the bearing at T, the nuts marked m being screwed into a head carried by the bars O, bringing the thrust plates at the point i. The reason for threading the nut m was to adjust it to the vanes in proper relative position to the sides of the pump. That is a simple construction maintained ever since, with the exception that ball bearings are now used.

8 Although the pumps were originally intended to take water and other loose material, such as sand and gravel, they proved capable of lifting practically anything that came in their way. The three following specimens are interesting as showing the pumps' lifting power:

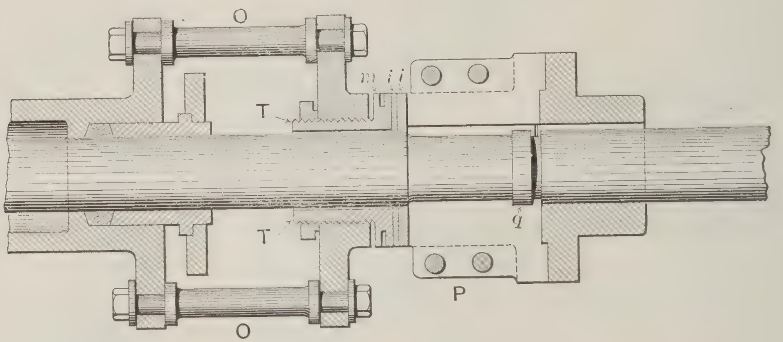

Fig. 4 Detail of Thrust Bearing of Pump

A piece of shaft weighing 70 lb. raised and passed by a 15-in. dredging pump; improvement of New York Harbor, Steamer Reliance.

A piece of tree root raised and passed by a 12-in. pump from 14 ft. of water at Miami, Fla.; Florida East Coast Railway Company improvements.

A piece of pig iron measuring $11\frac{1}{2}$ in. by $4\frac{3}{4}$ in. by $3\frac{1}{4}$ in. and weighing 35 lb. raised and passed by an 8-in. special cataract wrecking-pump from 15 ft. of water from the wreck of a canal boat sunk at Puas Dock, Yonkers, N. Y., by the Baxter Wrecking Company, New York.

9 For hydraulic dredging, the Government pays by the scow load and gets what is excavated. In ordinary hydraulic dredging, like that in the ship channel, about 15 per cent of the pump discharge was solid matter. About 40 per cent in excess of the amount deposited

in the bins went overboard with the overflow, and was carried out to the flats at the sides by the cross currents, which also carried the loose material stirred up by the drag. The result was that the Government obtained an excavation about 70 per cent in excess of what would have been obtained had all of the material removed from the bottom been caught in the bins. This, of course, greatly reduced the actual cost of the excavation. For example: the last contract made on the ship channel was at the rate of $16\frac{7}{8}$ cents per yard, while with the allowance indicated, above the actual cost per yard—channel measurement—it was about 11 cents.

10 As for the time of loading, some records indicate that this ship, 157 ft. long and with a capacity of 650 cu. yd., was loaded in 48 min.; there are also records of its being loaded at the rate of 16 cu. yd. per min., of solid matter placed in the bins; and records of its taking out to sea nearly 4000 cu. yd. per day. The vessel was worked in all kinds of weather, even when tackles had to be used to board her; and yet the ship was taking her load steadily. Except in the case of an actual breakdown the work could be carried on for 16 hr. per day.

John H. Norris. In a pumping plant of the character described, this type of equipment seems in the present state of the art the most suitable that could have been selected. I would like, in this connection, to call attention to another type of installation for service of this kind, though not on so large a scale, which appeals to me as being more desirable than the electric driven centrifugal pumping plant taking its power from the public utilities company.

2 At Coney Island was installed the first plant operated by the City of New York for fire protection by means of water delivered into mains under high pressure, with the idea of taking care of a restricted area where there was great danger from fire.

3 This plant consists of three 150-h.p. three-cylinder, vertical gas engines direct-connected to triplex pumps, each unit capable of pumping 1500 gal. per min. against a pressure of 150 lb. These engines take their fuel from the mains of the local gas company and can be arranged if necessary to run on gasolene. They are installed in a building on city property and are arranged to take their water supply from the city mains or from Coney Island Creek, within 50 ft. of the pumping station. The engines are started with compressed air, and the three units can be started up in less than three minutes. On every occasion they have been found ready for service whenever

the demand was made upon them. The cost of this pumping station was as follows:

Building	$10,000	
Equipment	37,000	
		$47,000

The annual operating expenses are:

Labor	$13,140.00
Supplies and Repairs	897.27
Fuel	150.00

4 By comparing the foregoing figures it will be evident that for service smaller than is required in the City of New York, the gas-engine-operated triplex pump gives an economical equipment that can be allowed to stand idle for any length of time and yet be ready for instant service.

5 New York City pays the New York Edison Company an annual charge of $90,000 for the privilege of calling for sufficient current to operate the equipment at any time. This item capitalized at 5 per cent would pay for a good-sized gas-engine plant.

6 The following data were taken from the capacity tests of the Coney Island units:

Duration of test	14	hr.
Average piston speed of pump	90.3	ft. per. min.
Total head pumped against	156.5	lb.
Average pump horsepower for each unit	142.2	h.p.
Average gas consumed per hour for the 3 units	8914.0	ft.
Average gallons per minute	4512.0	
Slip of pump	3.45	per cent
Average efficiency of pumps	82.00	per cent

J. R. Bibbins. Although Professor Carpenter's paper deals primarily with multistage pumps, I wish to direct attention to the question of motive power, upon which the success or failure of the system practically depends. We have seen excellent examples of two systems diametrically opposed in regard to power supply—the electrical and the gas-driven system. Under certain conditions, both are extremely serviceable. The first high-pressure installation on a large scale, in this country, was the gas-driven system at Philadelphia. Although I have not had an opportunity to follow the results of that station for the past two or three years, the results obtained and published for

the first year or so showed that such a system of gas-driven pumps merits every consideration.

2 First as to the security of power supply: In Philadelphia the Delaware Avenue station receives its gas supply directly from a 24-in. trunk main running between two very large gas holders, located in different parts of the city. Roughly, the pipe line measures four miles in length, its capacity constituting a considerable reserve in itself, if both the holders were unavailable. There is no intermediary apparatus whatever between the pipe line and the engine; that is the plant may draw directly on these two large holders of several million cubic feet capacity. This constitutes a very safe and reliable source of motive power which can hardly be paralleled except, perhaps, by the situation in the New York electric service, where there are so many stations to draw from.

3 In this connection, I would like to ask whether it is at present possible to utilize the storage battery capacity in the various substations for reserve service at the high-pressure pumping station? It is stated that the storage batteries are available for reserve in emergencies, such as discontinuance of the main high-tension current supply. I am under the impression that an inverted rotary requires a direct-driven exciter to maintain a definite frequency and prevent racing. Without special controlling apparatus, this inversion would be impossible in the ordinary sub-station equipment. Possibly special provision has been made in the New York systems, in which case, the security of power supply is certainly beyond criticism. In other words, would it be possible to invert the synchronous converters on short notice?

4 Second, quick starting: It seems to be a fact that a large part of the minimum time required for the starting of a fire-service station is consumed in the operation of the motor-driven by-pass valves. In Philadelphia these valves are operated from an independent supply, as in New York, and at least fifteen seconds are required to close them; whereas the engines are brought up to speed within half a minute from the time the signal is given, the remaining time being usually consumed in closing this motor-driven valve.

5 The various tests of the Philadelphia plant showed that each of the units could be readily put on the line in well under one minute. It is an interesting fact that the original underwriters' tests specified the time limit as twelve minutes for the starting of the first three units, whereas the whole station can be started in that time, and has been started in seven minutes.

6 During the 36 days of preliminary service trials of the Philadelphia station, out of one hundred alarms given, only four misses were made in getting any of the eleven units started. In not a single instance has the station, as a whole, failed to respond to the service, at least during the period over which my observation extended. This has been accomplished with the regular operating force of three men.

7 Third, in regard to the cost of service at Philadelphia: The only data on a large fire available, are those of the fire in the Coates Publishing House, which lasted about nineteen hours. The average cost for pumping was about six cents per thousand gallons, including gas, wages and supplies. The cost of the large East Side service, cited in the paper, is about nine cents for power alone, and I think this does not include the readiness-to-serve factor. On the other hand, it is patent that the cost of service in either the gas or the electrical station is relatively unimportant. The main desideratum is reliability.

8 Finally, I desire to advance an argument for the development of a new type of pump unit, namely, a high-speed gas-driven centrifugal pump. Some time ago, in connection with water-works service, I found great difficulty, even with the present high-speed single-acting gas engine, in matching engine speeds with those required in centrifugal pump work. However, for the pressure necessary in water-works practice, about 125 lb., one or two sizes of engines were found to be directly applicable to multistage pumps, with fair proportion of parts and good efficiencies. It seems possible to adopt a modified type of gas engine which would permit the direct connection mentioned.

9 This modification would naturally follow along lines of short stroke and high piston speeds with perhaps four cylinders. The engines at Philadelphia were designed with a piston speed of but 730 ft. per min. with a 22-in. stroke. This might be increased to 1000 ft. per min. without exceeding present-day limits, especially for units designed for occasional service. Such a unit would find immediate application in many industries and would combine the high economy of the gas engine with the simplicity of the centrifugal pump. The efficiencies shown by Professor Carpenter place the centrifugal pump in a position of closest competition with reciprocating pumping units.

J. J. Brown. I recently made a series of tests on three 6-in., 8-stage centrifugal pumps, each designed for 1000 gal. per min. and 560 lb. pressure at 1200 r.p.m. One of these pumps gave an efficiency from wire to water of 71 per cent, or a pump efficiency of 76 per cent. I

regret that Professor Carpenter did not give the results of his tests on the New York fire-service pumps at lower capacities. All of the tests were made at capacities considerably in excess of that for which the pumps were designed and they apparently show their best efficiency at approximately 25 per cent over the normal rating. This increased efficiency at excess capacity seems to be apparent in several recent tests made on high-lift centrifugal pumps. The 8-stage machines previously referred to give their best efficiency at 1300 gal., or about 30 per cent over rating.

2 Mr. White has raised a question as to the difference in efficiency between the New York fire-service pumps working in multiple and as separate units. I think this is occasioned by the variation in capacity of the pumps when working together on a common suction and discharge line. I have found it rather difficult to balance two centrifugal pumps on a common discharge, and pitot tube tests indicate in almost every case a considerable difference between the amounts of water handled by the individual units under these conditions.

3 I have in mind one installation on fire service, where the pumps were called upon to deliver against the maximum pressure for which they were designed and it was only with considerable difficulty that we were able to cut in additional units. I think that if venturi meters or pitot tubes had been placed on the discharge of each of the five pumps when they were working in multiple, a difference in capacity of the several units would have been shown, which would account for the difference in efficiency observed when the pumps were working individually and not in multiple.

George A. Orrok. At the time of the award of contract for these fire pumps, the New York Edison Company was obtaining proposals for centrifugal feed pumps—a somewhat similar service—and eight 1000-gal. 300-lb. pressure fire-stage pumps were purchased. There was no attempt to obtain a high guarantee for efficiency, but the builders did state that under the above conditions an efficiency of 65 to 68 per cent would be obtained. These pumps were of the Jager type and under test showed an efficiency of about 68 per cent.

2 Fig. 5 shows that the high-pressure fire-service pumps are of the Kugel-Gelpke type and should be a trifle more efficient because of smaller friction and leakage. Seventy-one per cent seemed a very high efficiency and many doubts were expressed regarding the fulfillment of the guarantees. The extreme figure of 79 per cent obtained is probably the result of careful design and extra good shop

work and I believe has not been excelled. That this figure came as a surprise may be explained by the fact that most centrifugal pumps are stock pumps and not specially designed for the work they have to do. Pump manufacturers have been more concerned in getting a line of patterns that will suit standard conditions than in developing a line of pumps and system of patterns capable of doing the best work.

3 As a centrifugal pump is a reversed mixed-flow or Francis reaction turbine, similar care in design and construction would probably give efficiencies similar to those of the best makes of reaction turbines, which approximate 90 per cent.

Frederick Ray. The difference in efficiency of the units operated individually from that obtained when several were operated in parallel might be due to the different rates of flow through the venturi meters under the two conditions. With one pump operating, this flow would be low and the mercury column reading would be but slightly over an inch, so that with a given error of observation the percentage of error would be much greater than with two or three pumps discharging through the same meter.

2 Professor Carpenter here replying that the pipe connecting the two meters was open all the time, Mr. Ray continuing said:

3 This would equalize the flow in the meters when the whole station was running, so that the mercury column reading would be about 6¼ times the reading with one pump. It has not been my experience that parallel operation of a number of pumps has any tendency to decrease or otherwise change the efficiency obtained when operated individually. The efficiency should be the same, and in this case, as the pressures were taken at each pump, any losses in the piping system due to parallel operation would be external to the gages and would not show in the calculations. If the pressure had been taken at the discharge of the whole system, losses in the piping would affect the results.

4 Many pumps are running under similar conditions, at the efficiencies given. I have myself obtained efficiencies of 79 or 80 per cent and higher, but I do not rely as much on them as on some a little lower. I am now testing a 6-in., 2-stage underwriter pump, having a normal capacity of 500 gal. per min. against 100 lb. pressure, which has developed a maximum efficiency of 73 per cent.

5 I think the centrifugal pump is the ideal one for fire service, not only on account of its simplicity and reliability, but also on account of its characteristic increase in capacity as the pressure is

reduced. Thus, the 500-gal. underwriter pump referred to will discharge 870 gal. per min. at 60 lb., or enough for four streams at this pressure. It will give three streams at 90 lb., two streams at 110 lb. and one at 117 lb.—all at constant speed without any regulation whatever.

6 The City of Toronto has recently issued specifications for centrifugal pumps for their general municipal water supply, among which are several fire pumps capable of discharging against 300 lb. pressure. These pumps, however, are to be equipped with variable-speed induction motors, the pressure regulation being obtained by speed variation. This is superior to throttling regulation from the standpoint of current economy and in the case of the New York installation a considerable saving could be made by this means, as most of the fires can be handled with 200 lb. pressure or less.

H. Y. Haden. A somewhat unusual result is obtained from this type of pump, for as the total head continues to increase beyond a certain point, the capacity falls off, with the result that the capacity curve, as given in Fig. 7, shows a backward tendency. It will be interesting to get the explanation of this.

2 There is unquestionably a large field in fire protection for steam-turbine-driven centrifugal pumps, and it is to be hoped that the Fire Underwriters will officially accept this type of fire protection unit. I believe that a properly designed centrifugal pump, for high speeds and of few stages, can be used to great advantage when direct-connected to high-speed turbines.

Thomas J. Gannon.[1] It was decided to use electricity as power for the pumping stations, because the first cost of installation and the yearly cost of operation and maintenance and fixed charges were estimated to be lower, taking into account the intermittent service. The construction and operation of a steam plant were entirely out of consideration and the choice lay between gas-engine-driven and electric-driven pumps receiving power from outside sources.

2 It was estimated that gas operation of plants equal in capacity to the present electrically driven plants, would involve a fixed charge of $50,000 a year, in addition to the cost of the gas actually consumed. The question as to who should build and maintain

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough, New York.

the necessary large gas mains, the cost of which would approximate a million dollars, was not definitely settled. That the cost of a gas-engine-driven pumping plant would have been approximately double, both for machinery, building and area of land to be purchased, is borne out by the actual cost of the installations in Manhattan and at Coney Island.

3 The capacity of the gas-operated Coney Island plant is 4500 gal. of water per min. against a head of 150 lb. per sq. in. The cost of the machinery is approximately $37,000 and the cost of the building approximately $10,000. The combined capacity of the two pumping plants in the Borough of Manhattan, as originally laid out, was 30,000 gal. per min. against a head of 300 lb., with provision in each station for three additional pumping units of a capacity of 3000 gal. each, making a total combined capacity of 48,000 gal. per min. against 300 lb. pressure. On actual test, however, the capacity of the pumps was approximately 20 per cent greater than the designed capacity.

4 Furthermore, the flexibility of this type of pump permits of an increased discharge at lower pressures, which gives a capacity of approximately 5500 to 5600 gal. per min. for pressures between 150 and 200 lb., or a combined total capacity of 55,000 gal. per min. against 200 lb. pressure. This corresponds to the pressure at which the station is operated for most fires. In other words, the water horsepower of one plant, as compared with the other, is approximately in the ratio of 20 to 1.

5 The first cost of installation of the gas-engine-driven plant is therefore more than double the first cost of installation of an electrically driven plant, in the city of New York. The cost of each of the two Manhattan pumping stations complete, exclusive of land, was practically $240,000.

6 The high-pressure fire-service pumping stations went into official operation on July 6, 1908. It was at first decided to put the stations in service only when called on by the fire department, and up to and including November 20, 1908, the pumping stations were called upon to go into actual service for but 17 fires. On that date, the method of operation was amended so that the pumping stations are put in service in response to every alarm in the high-pressure district, and continue in operation awaiting instructions from the fire department. Under this system, from November 20 to December 31, 1908, the pumps responded to 116 first alarms. From the best available information, water was used in 55 instances, making a

total of 72 fires for which the high-pressure service had been used up to that date.

7 To insure readiness for service at all times, daily tests are made, of at least half an hour's duration, unless the station has been in actual operation during the preceding 24 hours.

8 During the first quarter of 1909 the number of alarms received was 239, and water was taken from the station for 125 actual fires. The total amount of water pumped was 17,840,000 gal., and 145,900 kw-hr. was consumed. It was on January 7, 8 and 9 of this quarter that the three large simultaneous fires mentioned in Par. 75, occurred, for which over 14,000,000 gal. of water was pumped, leaving about 3,800,000 gal. for the balance of actual fires occurring during the quarter. For these three simultaneous fires more than 81,000 kw-hr. was consumed while the total consumption of power for the quarter for all fires and testing purposes was but 145,900 kw-hr.

9 As to why a pump running singly develops a higher efficiency than when running in conjunction with several others, it is observed that pumps of the same type do not necessarily develop their best efficiency at the same speed and pressure. The pump running singly will naturally develop a pressure which corresponds to its own design, but when working in multiple, it will have to adjust itself to the common pressure.

10 As to reliability I have neither seen nor heard of any time when any one of the ten pumps installed in the Borough of Manhattan has failed to respond instantly when called on for service and to develop the full pressure on the system within one minute's time. At no time in service have the pumps shut down of their own accord.

Henry B. Machen.[1] Among the many difficulties encountered during the construction of the distribution system, perhaps the greatest was that due to the congested sub-surface of the street, which was a source of continual extra expense to the contractor, and of worry to the man in charge of selecting the location for the excavation of the trench.

2 The intersection of Sixth Avenue and Fourteenth Street may be cited as an example, since complete notes are available, due to the station excavation for the Hudson Tunnels. Here there were nine gas mains east and west, and nine north and south, belonging to

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough, New York.

four different companies; two water mains in each direction; sewers and their connections on each side of the street; five Edison duct lines, and five duct lines with large manholes belonging to the Consolidated Telegraph Subway Company or the Empire City Subway Company; the conduits and banks of ducts of the Fourteenth Street and the Sixth Avenue trolleys; and lastly, the columns of the elevated railroad with their deep foundations.

3 Through this network the high-pressure main had to be so laid that the construction of the Sixth Avenue tunnel would not require it to be relaid. The excavation was carried on by tunneling, with here and there an opening through which the earth could be hoisted, using a pail let down by a rope. The pipe was lowered into the trench some distance up the street and pulled through, piece by piece, inspection of the running of the joint and caulking being almost impossible, since the space admitted but one man at a time after the pipe had been hauled in.

4 This condition existed at nearly all intersections of the main thoroughfares, such as Broadway, Sixth Avenue, Fifth Avenue, the Bowery, etc., and accounts for the high cost of laying the mains, averaging about $11 per ft. complete.

5 The second great difficulty encountered was in obtaining the prescribed test, which called for 450 lb. pressure per sq. in. to be held for 10 min., during which time the leakage was measured.

6 The system contained about 40,000 castings, 30,000 being straight pipe, tested at the foundry to 650 lb. The specials were not tested. All these castings, as already stated, were tested in the ground to 450 lb., the mains being under pressure in sections about one block long, between gates.

7 During the eighteen months the system has been in service, there have been but three breaks in the mains, all three in castings which had been subjected to the foundry test of 650 lb., two breaking at 150 lb. and the third at 300 lb. pressure.

8 To overcome the danger should a break occur during a fire, the proposed extensions to the distribution system now under contract, amounting to about $1,500,000, are laid out on what the department calls the duplex system. This method of overcoming the difficulty was first suggested by Mr. Blatt, assistant engineer of the High-Pressure Bureau. It consists of laying two entirely independent systems of mains and hydrants in alternate streets, the hydrants of one system being painted red and the other green. The mains are so laid out that at nearly all intersections of streets hydrants of both colors are available.

9 Should a break occur in either system, the operator at the pumping station would at once know in which system the trouble was located by looking at the venturi meters, and by throwing a switch he would start the closing of two electrically driven valves, separating one system from the other. Hydrants would then be available and in service pending the location and isolation of the damaged section.

10 The section now in operation was designed to give 20,000 gal. per min. on any one block with a loss due to friction from pumps to hydrant not to exceed 40 lb. The duplex extension will give the same results, and should either half be out of service by an accident, there will still be available at the same location 10,000 gal. per min., with a loss from the pumps to the hydrant in the most unfavorable location not exceeding 50 lb.

Richard H. Rice. This paper shows that the installation described has been made after the most careful study and a very intelligent choice of the types of apparatus to be used. The choice of the centrifugal pump for the work described is thoroughly justified by its simplicity and by the efficiencies obtained. The choice of alternating current as the source of power, in view of the unlimited supply of current existing and the duplicate means of conducting it into the station, is also justified. The centrifugal pump is today the popular means of producing pressure for emergency fire purposes, as in the fire boats of New York, Chicago, Duluth and San Francisco, and the new high-pressure service of San Francisco. In San Francisco twelve of these pumps are now being installed, four on fire boats and eight for an auxiliary fire installation. On the fire boats centrifugal pumps are particularly adaptable as they can be run in series or in parallel. In parallel they give 150 lb. pressure, and in series the pressure is doubled. This pressure is particularly valuable where walls have to be battered down, or streams thrown long distances.

2 In cases where electricity is not so available as it is in New York, steam turbines are being installed, and they offer advantages over the gas engine, where maximum reliability is considered.

3 As an emergency installation pure and simple, I think the installation mentioned in the paper can be still further simplified. I believe the speeds chosen for operating the pumps are too low, and that the pumps contain too many stages. I have had occasion to make extensive researches in centrifugal pump design with special reference to operation at steam-turbine speeds, and have found that

they can be operated at high speeds with a smaller number of stages, giving efficiencies comparable with those obtained here, although the question of efficiency is subsidiary to reliability for this service. Pumps for this service should be designed with two or three stages at the most, and with considerably higher speed.

4 Pumps can also be designed without balancing pistons, which are undesirable from the viewpoint of possible interruption of service. An inspection of Fig. 5, illustrating the construction of the pumps, will show that the balancing pistons used are quite liable to damage if water containing sand or other impurities is used, and this damage would very probably result in stoppage of the pump when it is badly needed. The use of balancing pistons is unnecessary in such emergency apparatus and should be avoided.

C. A. Hague. A question has been asked several times with reference to the results of tests of efficiency on centrifugal pumps operating singly and in multiple or group. Professor Carpenter has given the very plausible explanation that the difference in efficiency in favor of the pumps running singly is probably due to the presence of eddies and disturbances in the pipes when the pumps are operating together and the absence of such eddies and disturbances when only one pump is at work. In my experience in installing pumps and condensers singly and in groups I have found them extremely sensitive to each other in operation, both in taking in and discharging the water, when more than one pump is working on a line.

2 In the Manhattan stations, it seems to me that the suction or inlet pipes and the discharge pipes are coupled too closely for best efficiency; and also that the inlet pipe close to the pumps is not large enough for operation in multiple, although perhaps ample for a single pump when the water is undisturbed by the draft and discharge of several pumps. I have experimented considerably in that line, and have found that a comparatively large body of water next to the pumps on the suction side will materially ease the machines in their performance. The idea is to come up to the building with a normal supply pipe, and then enlarge it very considerably just where it enters the building, providing the inlet pipe with a good-sized air chamber wherever possible. I have tried this several times with excellent results.

3 Mr. Brown mentioned the difficulty of cutting in with a second pump where the first pump was already running, a difficulty which

I think is also due to too close connections along the inlet and outlet lines and a cramped condition generally. Of course, a disturbance in the water column and in the hydraulic horsepower would unbalance the electric power to a certain extent, perhaps not much, but the total disturbance may very easily result in the loss of several points in the efficiency.

4 Considering the fact that the city pays by the kilowatt-hour for its electric current as per switchboard reading, it would be no more than proper to state the efficiency of the machine as a whole, and not exclusively upon the basis of motor efficiency obtained in the shop of the makers a thousand miles or so away. In this case when 100 h.p. in current is supplied to the switchboard, the motor has shown an output by a competent test of 93.2 h.p.—Par. 37—the 6.8 h.p., although charged against the city in the power bills, being lost in heat and friction. Then, all that is charged against the pump is 93.2 h.p. The 67.57 h.p. shown by the pump for each 100 h.p. at the switchboard indicates only 67.57 per cent total efficiency, although the 67.57 h.p. indicates 72.5 per cent efficiency of the power delivered by the motor. I have tested several centrifugal pumping plants of various sizes and powers, and the total efficiency generally shows from 64.5 per cent to about 68 per cent and very seldom above the latter figure.

5 Mr. Bibbins touched upon the possibilities of utilizing the centrifugal pump for waterworks service, but upon investigation he would find a vast difference between emergency service, where operating economy counts for little in the face of great danger from fire, and the steady and necessarily economical service required for the continual pumping in waterworks stations. To show how deceptive a portion of the truth may be, a case is cited where a pumpage of a capacity of 10,000,000 gal. per day against 110 lb. load could easily be accomplished with displacement steam machinery by an expenditure of $10,000 per annum for coal. But an attempt to drive centrifugal pumps by electricity resulted in a cost for electrical power, at $6.50 per 1,000,000 gal., of $23,725 per annum; showing a difference in favor of displacement steam machinery that would pay 5 per cent per annum on $275,940. There is no conceivable difference in cost of machinery, buildings, maintenance, attendance, or anything else, that would justify such a preference for electricity and centrifugal pumps over steam and displacement pumps. Note the following figures:

10,000,000 gal. daily, against 110 lb.	440 pump-h.p.
120,000,000 steam duty with 8 lb. evaporation in the boilers, coal at $2.50 per net ton delivered	$9928 per annum
Electric power at $6.50 per 1,000,000 gal. against 110 lb. means 3,650,000,000 gal. per annum at $6.50	$23,725 per annum
The difference in cost for the element of power is $13,797 per annum, which at 5 per cent would capitalize at	$275,940

6 The steam-driven, reciprocating, displacement pumping engine can show a mechanical efficiency, from the power put in through the throttle, to the water-horsepower of the pumps, as high as 96 per cent, never as low as 90 per cent, under the above conditions. The centrifugal pump when steam-driven has a corresponding efficiency of about 65 per cent, and when electrically driven of about 67 per cent. A comparison of tests is given in the tables, in which it will be seen that the steam plant saves enough to pay 8.6 per cent on its entire cost.

TABLE 1 COST OF OWNING AND PUMPING WITH HIGHEST TYPE AND CLASS OF STEAM PUMPING MACHINERY

One Unit, Steam-Driven, Reciprocating, Displacement Machinery, Capacity of 25,000,000 Gal. Against 87 Lb.

Pump horsepower	870
Boiler horsepower for triple-expansion vertical pumping engine	450
Engine house and foundations and engine foundations	$150,000
Boiler house and foundation, boiler foundations, chimney, etc	
Vertical triple-expansion pumping engine	
450 h.p. of boilers	
Building for coal supply	

CHARGES AGAINST PLANT—PUMPING ENGINE

Interest	4 per cent
Sinking fund	5 per cent
Depreciation	2 per cent
Oil waste, etc	1 per cent
Total	12 per cent

CHARGES AGAINST PLANT—BOILERS

Interest	4 per cent
Sinking fund	5 per cent
Depreciation	5 per cent
Total	14 per cent

3 engineers. 6 firemen. 3 oilers.

Coal at $2.10 per net ton

SUMMARY FOR STEAM RECIPROCATING MACHINERY

Coal per annum	$11,957.40
Wages per annum	9,900.00
Capital charges on engine	13,920.00
Capital charges on boilers	1,260.00
Capital charges on buildings	1,548.00
Total charges per annum	$38,585.40
Cost per 1,000,000 gal	$4.11
Cost per horsepower	43.16

TABLE 2 COST OF OWNING AND PUMPING WITH HIGHEST TYPE ELECTRO-TURBINE PUMPING MACHINERY

ONE UNIT, ELECTRIC-DRIVEN, CENTRIFUGAL MACHINERY, CAPACITY 25,000,000 GAL. AGAINST 87 LB.

Pump horsepower	870
Two-stage, electric-driven centrifugal pump Engine house and foundations and pump foundations Transformer house and foundations Transformers, lightning arresters, conductors, controllers and auxiliaries	$43,750

CHARGES AGAINST PLANT—PUMPING MACHINERY, ETC.

Interest	4 per cent
Sinking fund	5 per cent
Oil, waste, etc	1 per cent
Depreciation	2 per cent
Total	12 per cent

3 Engineers. 3 Extra men
Electric current, $4.50 per 1,000,000 gal.

SUMMARY FOR ELECTRIC-TURBINE MACHINERY

Electric current per annum	$41,062.50
Wages per annum	5,700.00
Capital charges on machinery	4,314.00
Capital charges on buildings	468.00
Total charges per annum	$51,544.50
Cost per 1,000,000 gal	$5.64
Cost per horse power	59.24

Thos. J. Gannon. In reply to Mr. Hague I will read the conditions which occurred on the evening of January 7, when both pumping stations were put to a crucial test:

7.22	First alarm, Hudson and Franklin Sts.
7.28	Second alarm, Hudson and Franklin Sts.
7.29	Third alarm, Hudson and Franklin Sts.
7.46	Fourth alarm, Hudson and Franklin Sts.
7.54	First alarm, Bowery and Hester Sts.
8.17	Automatic, Mercer and Houston Sts.
8.19	Second alarm, Bowery and Hester Sts.
8.29	First alarm, Mercer and Houston Sts.
8.32	Third alarm, Bowery and Hester Sts.
8.40	Second alarm, Mercer and Houston Sts.
8.43	Third alarm, Mercer and Houston Sts.
8.45	Fifth alarm, Mercer and Houston Sts.

2 In due time seven pumps were put into operation, with a discharge which reached at times over 35,000 gal. per min., and it was estimated that over 52 fire streams were in service at the same time. Each pump responded instantly and remained in service until ordered shut down. The pressure was ordered gradually increased from 125 lb. to 230 lb., where it was maintained throughout the greater part of the time that the fires raged. The operating force at each pumping station consisted of but one engineman, one oiler, one telephone operator and one laborer.

Prof. George F. Sever. A question was asked as to the feasibility of using the storage battery capacity to invert the rotaries and provide alternating current, to be spread through the alternating-current system to the sub-stations, and from those to provide alternating current to the pumping stations. In our preliminary investigation, if I recall the facts correctly, we were assured that this could be done; giving us another feature of reliability in the operation of the system. If the Waterside station should go out of business, we could still get current from the sub-station.

A. C. Paulsmeier.[1] While the reasons given in the paper for the selection of electric-driven turbine pumps do not coincide with the conclusions as to reliability that have been reached in the West, there can be no question about the careful study given by the engineers who planned the high-pressure fire system described.

[1] Chief Engineer, Byron Jackson Iron Works, San Franciso, Cal.

2 The pumps show a remarkable efficiency, and one of the principal points that should commend them to those interested is their great flexibility as to capacity, a characteristic that every fire pump should possess.

3 The eight fire pumps now being built for the City of San Francisco are of a combined capacity of 216,000 gal. per min., under a working pressure of 300 lb. Each of these pumps is driven by a 750-h.p. Curtis steam turbine, operating at a normal speed of 1800 r.p.m.

4 In addition there are now being completed four fire pumps for the boats Dennis Sullivan and David Scannel, of an aggregate capacity of 9000 gal. per min. under 300 lb. working pressure, or 18,000 gal. per min. under 150 lb. working pressure, the pumps being so arranged that they work either in series or in parallel. The pumps have all been subjected to 24-hr. tests, and while the data on these tests are not sufficiently complete for publication, it was shown that the pumps are not as flexible as to capacity, or are not as capable of pumping an excess quantity of water, as are the Manhattan pumps. The reason for this is that the impellers in the San Francisco pumps are only $13\frac{5}{8}$ in. in diameter, while the inlet to the impellers is less than 10 in. in diameter, this opening being further restricted by the pump shaft, so that it is impossible to obtain much excess water from these pumps, no matter how much below the normal the discharge pressure is carried.

5 In the station pumps now being built the velocities at the entrance to the impellers have been somewhat decreased, although it is impossible to make anything like the excess capacity shown by the Manhattan pumps, which have impellers of such a size that the inlets may be made anything consistent with good practice.

W. B. Gregory. It is gratifying to know that efficiencies ranging from 70 to 80 per cent may be obtained with well designed five-stage turbine pumps. The high-pressure fire-service pumps in New York represent one extreme of conditions, while at the other extreme is the centrifugal pump used in the rice irrigation territory of Louisiana and Texas for raising large quantities of water through comparatively small lifts.

2 The improvement in design of pumps of the latter class in the last ten years, and especially in the last five years, has made it possible to specify an efficiency of 75 per cent, even with heads as low as 10 ft. Purchasers of pumping plants in this section are no

longer satisfied with pumping outfits having efficiencies ranging from 50 to 60 per cent.

3 As examples of the results obtained with pumps of the class that deals with large volumes of water, the tables are quoted from recent acceptance tests conducted by the writer, of pumping plants used for rice irrigation.

TABLE 1 ACCEPTANCE TESTS

TANDEM-COMPOUND CONDENSING ENGINES, DIRECT-CONNECTED

Cane and Rice Belt Irrigating Company, Fulshear, Texas, August 12 and 14, 1908

WORTHINGTON PUMPS	FIRST LIFT	SECOND LIFT
Size of pump (diameter discharge pipe), in	45	45
Water pumped, gal. per min	47,620	46,430
Head on pump, ft	33.90	13.95
Efficiency of engine and pump, %	69.5	73.6
Efficiency of pump (engine 93 %)	74.7	79.2

CROSS-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Sabine Canal Company, Vinton, La., May 22, 1909

WORTHINGTON PUMP	
Size of pump (diameter discharge pipe), in	45
Water pumped, gal. per min	44,010
Head on pump, ft	23.26
Efficiency of engine and pump, %	69.5
Efficiency of pump (engine 90 %)	77.3

TANDEM-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Second Lift, Neches Canal, July 16, 1909

MORRIS MACHINE WORKS PUMP	
Size of pump (diameter of discharge pipe), in	48
Water pumped, gal. per min	60,300
Head on pump, ft	10 12
Efficiency of engine and pump (maximum), %	69.9
Efficiency of pump (engine efficiency 93.2 % max.)	75

CHARLES B. REARICK. Electrically driven fire pumping-stations for large cities are dependent upon current from an outside source. usually a large central power plant. It would seem quite practicable in many cases to locate new fire pumping stations adjacent to some large power plant having considerable boiler capacity. In such cases it would be possible to drive the centrifugal or turbine pumps with steam turbines, and thus eliminate the necessity of large over-

load capacity in electric generating units for the central station, and also the liability of derangement of the lines between the power stations and the pumping stations. The charge for standby service per annum should be less than for similar electric service.

2 The steam turbines have the advantage of being operative at any speed, and in this manner will maintain in the discharge mains any pressure desired. Furthermore, automatic regulating valves can be used in connection with the turbine to maintain constant pressure irrespective of demand or flow.

3 It is probable that the cost of installation would be less than for electric-driven units. The turbine could run non-condensing, as the question of steam consumption is of small moment for fire service.

Henry E. Longwell. The last paragraph of the paper furnishes a striking illustration of how purely academic is the ordinary official efficiency test, and how valueless it is as a basis on which to predicate the results that may be expected when the plant is operated under normal service conditions.

2 The average net pressure against which the 14,095,000 gal. was pumped with a current consumption of 81,450 kw-hr. is not stated. Assuming that it was 300 lb. net per sq. in., the pump efficiency, after allowing for the losses in the motor, would be only 40 per cent. However we know that for part of the time the pressure did not exceed 225 lb., or, considering the pressure in the suction mains, about 200 lb. net. If the entire quantity of water had been pumped against this lower pressure, the efficiency would be well under 30 per cent. It is therefore perhaps fair to assume that the actual average efficiency was not far from 35 or 36 per cent, or say, in round numbers, only one-half that reported as shown on the official test.

W. M. Fleming. With the rapidly increasing size and height of office buildings, the annual fire loss in the business districts of the cities of the United States is increasing to an alarming extent. The installation of these tremendously effective fire-fighting systems has already proved of definite value in the reduction of city fire losses, and consequently of insurance costs.

2 What was probably the pioneer large and independent so-called high-pressure fire system in this country was installed at Philadelphia in 1903-1904. This plant differs in almost every important detail from the New York system more recently installed;

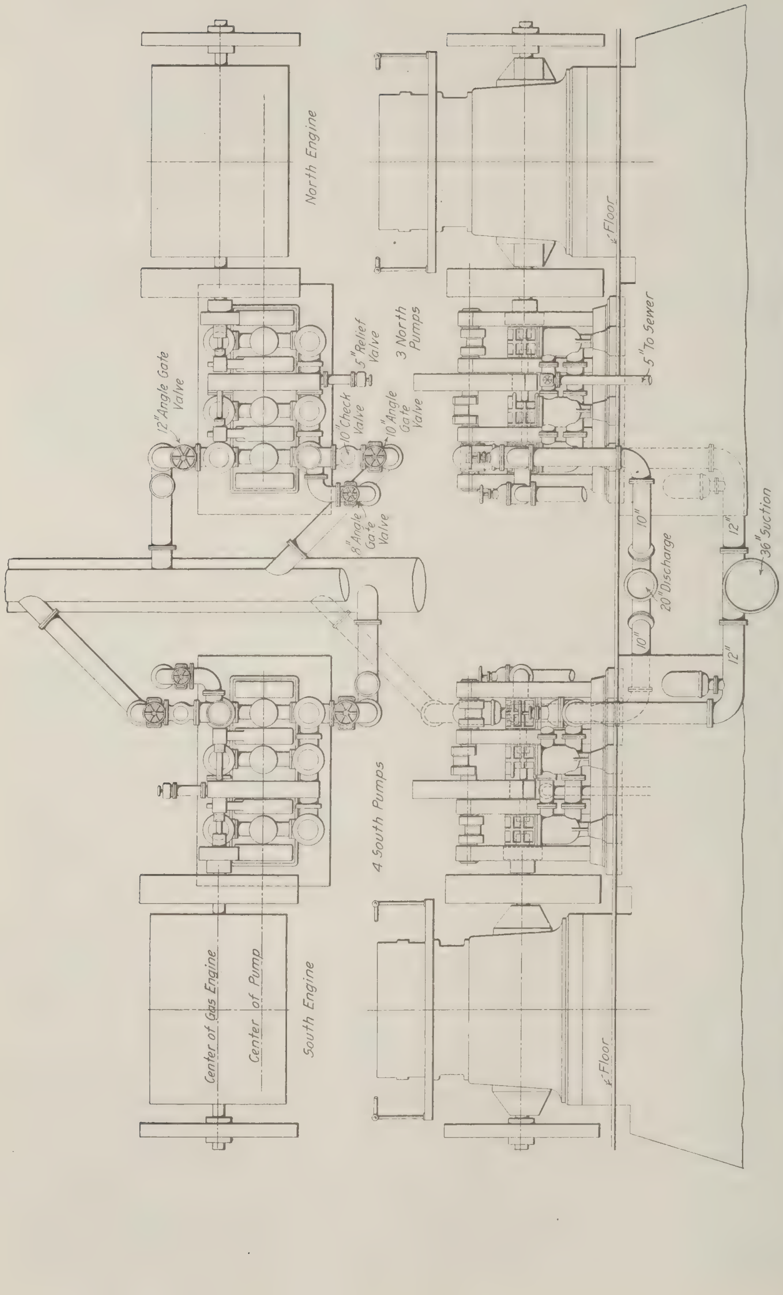

FIG. 1 GENERAL ARRANGEMENT OF THE PHILADELPHIA HIGH-PRESSURE FIRE-PUMPING STATION

yet the general results in both cases have been excellent. In Philadelphia the plant has so many times proved of great value in actual service that a much larger fire-fighting system, consisting of pumping units identical with those originally selected, is now being installed to protect what is known as the Kensington mill district.

3 From the original Philadelphia station at Delaware Ave. and Race St., a location unlikely to be seriously injured by conflagration, Delaware River water is supplied to independent high-pressure fire-service mains which effectually cover more than 425 acres at the center of the business district. The pumping units consist of vertical double-acting triplex power pumps built by the Deane Steam Pump Company, direct-connected to Westinghouse

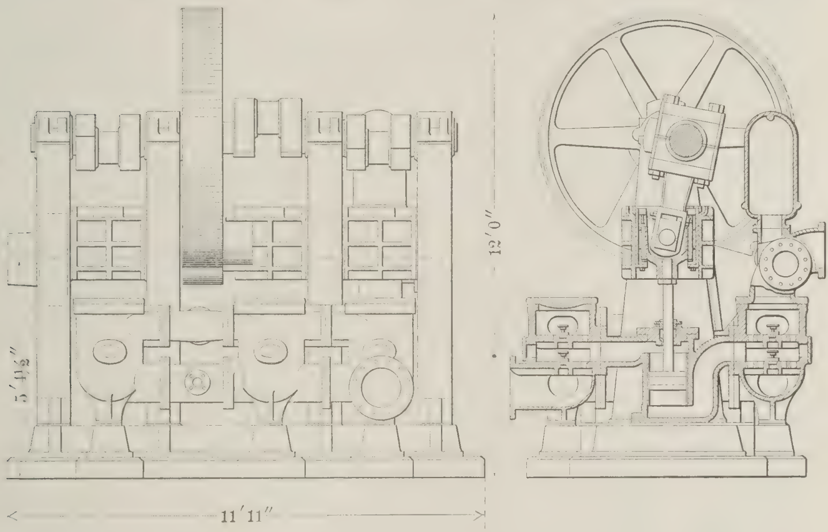

Fig. 2 Side and Sectional End Elevation of Triplex Pumps for the Philadelphia High-Pressure Fire-Pumping Station

vertical 3-cylinder 4-cycle gas engines each of 280 h.p. The seven large pumping units have each a nominal capacity of 1200 U. S. gal. per min., at 300 lb. pressure, and two small units have a capacity of 350 U. S. gal. at the same pressure.

4 The general arrangement of the Philadelphia pumping station is similar to that of the large New York installations. (See Fig. 1.) Two rows of pumping units occupy the main floor of the station. The pumps are nearest the center, and the gas engines are located in the same relative positions thereto as the motors in the New York

pump houses. A platform extending along the sides of the building, about ten feet above the floor, serves as a working gallery for the operation of the engine throttles. Space is provided for the installation of three additional pumping units, and all mains are proportioned with the ultimate probable capacity of the plant in view. Suitable connections are provided to the mains so that the capacity of the pumping station may be supplemented by the use of the city's powerful fire boats, should occasion require.

5 The internal combustion engines are of the well known standard Westinghouse type and require little explanation. Speed regulation with varying loads is accomplished by the action of a centrifugal governor controlling the quantity of combustible admitted to the cylinders. Ignition is by a very neat type of make and break mechanism contained in a cylindrical plug. Two independent igniters are provided in each cylinder, and three independent sources of ignition current are available at all times. The engines are started by the use of compressed air, which is admitted to one of the cylinders at the proper time to secure rotation in the direction required until the regular cycle of operation is established. The pumps are started under no-load.

6 The pumps are of the vertical, double-acting piston, triplex power type, requiring comparatively small floor space and giving a rate of discharge so smooth and uniform as to make imperceptible at the hose nozzles any pulsation in pressure.

7 Fig. 2 is a sectional view of one of the pumps, indicating quite clearly the extreme simplicity and accessibility of the machine, and its general construction. All valves are of the poppet type, readily accessible through handhole openings. Valve areas and waterways naturally are comparatively large, so that friction losses are reduced to a minimum. The water ends are thoroughly brass-fitted in order that the pumps may be readily started after a long period of disuse.

8 There is a connection through a 12-in. check valve, from the city mains to the high-pressure system, so that the mains and pumps are constantly primed with a pressure of 60 lb. and are ready for service at all times. A complete system of fire-alarm boxes and telephones, with underground wires, permits direct communication between the vicinity of any fire and the pumping station. On the sounding of the alarm, the station force, consisting of an engineer and his assistant, can bring the total plant of seven large units into service in seven minutes, and have repeatedly done so. Work-

ing pressure is invariably available at the hydrants one minute from the time of the alarm. Such a result would be impossible with ordinary movable apparatus.

9 The pumping units are started up under no-load, by the use of a motor-driven by-pass valve, through which the pump discharges into an overflow, until the normal cycle of operations has been set up in the gas engine, when the switch is closed, causing the by-pass valve to close and the discharge to be directed into the fire mains.

10 Experience has indicated that the maximum pressure of 300 lb. is required only for the most extensive fires, and for fires in the higher parts of tall buildings. The pressure records show that probably 75 per cent of the water pumped is required at not more than 150 lb. to 175 lb. pressure. The pressure desired in each case, is dictated over the telephone by the fire chief, the required pressure regulation being obtained by proportioning the number of units in operation to the requirements.

11 The practical results of the use of the Philadelphia fire system have been: material reduction in fire losses in the protected district, large decrease in fire insurance rates, and a greater willingness on the part of property owners in the protected section to erect pretentious office buildings.

12 Though the writer is unable to present a statement as to the annual saving to property owners by the installation, yet in view of the low cost of operation of the plant, there can be no question but that it presents a considerable yearly saving to the city. During the year 1907, which is perhaps typical, water was delivered to 16 fires, the longest one lasting 44 hr. The plant responded to 116 alarms at which no service was required. The operating expenses for the year were as follows:

Gas, 839,488 cu. ft. at $1.00	$839.49
Electric lighting	343.99
Electric power	7.98
65 tons pea coal at $3.50	277.50
Supplies furnished the pumping station for the entire year 1907	1,500.00
Total fixed charges for 1907	$2,968.96

Summary

Salaries (Total for entire staff)	$8,389.72
Total cost materials	2,968.96
Total operating expenses	$11,358.68
Total daily maintenance charge, salaries and operation	$31.12

13 No mechanical defects have yet developed in either engines or pumps, and practically the only replacements have been a few rubber valves for the pumps and ignition details for the engines.

14 While no definite comparison can be made between the small plant in Philadelphia and the comparatively large plants in New York, which have not yet been in operation for an appreciable length of time, the operating expenses of the Philadelphia plant seem likely to prove much less for a given quantity of service. This is largely due to the so-called "readiness-to-serve" charge made by the company furnishing power to the New York plants. To this charge must, of course, be added the actual cost of the current consumed.

15 Unfortunately no mechanical efficiency test has ever been made on any of the Philadelphia pumping units. Judging from tests of similar machinery, an efficiency of 80 to 85 per cent is to be expected from pumps of this character operating against 150 to 200 lb. pressure. If this is the case, knowing that 75 to 80 per cent of the water to be used will be required at pressures not to exceed 175 lb., it would seem that the plant efficiency in Philadelphia would prove greater than in New York, where we understand that the water must be delivered through reducing valves from 300 lb. to any lower pressure required.

Note: The discussion of this paper at St. Louis will be published later, with the author's closure.

STRESSES IN REINFORCED CONCRETE BEAMS

By Prof. Gaetano Lanza and Lawrence F. Smith,[1] Published in The Journal for Mid-October

ABSTRACT OF PAPER

This paper presents a comparison, in the case of eleven beams, of (*a*) the position of the neutral axis, (*b*) the stress in the steel, (*c*) the greatest compressive stress in the concrete, (*d*) the greatest deflection of the beam as determined by experiment, with the same quantities as computed by each of three well-known theories of the distribution of the stresses, these theories being designated by *A*, *B* and *C* respectively. *A* and *B* both neglect the tension in the concrete, but differ in the mode of distribution of the compression; while *C* takes account of tension in the concrete. The results show a better agreement with the results of the experiments when tension in the concrete is considered than when it is not. This is especially so in the case of the stress in the steel and in the position of the neutral axis, and, to a lesser degree, in the greatest fibre stress in the concrete and in the deflection.

DISCUSSION AT BOSTON

Chas. T. Main. All engineers, civil, mechanical or any other, want to know the most accurate way of figuring the stresses in reinforced concrete. What I am more anxious to know is that the proper ingredients are used, with proper mixing and good workmanship, so that we may be reasonably sure of a factor of safety in the finished work somewhere near what was intended. I have done no work of this sort without constant supervision, and am obliged to say that I have done no work that has been a source of pleasure to me. All of the building materials in common use are, I think, more certain in results than reinforced concrete. It is quite necessary to improve in the use of this material and in workmanship, in order to produce work which will inspire confidence.

[1] Instructor, Mass. Inst. of Tech.

Sanford E. Thompson. Professor Lanza's paper is of much value as a means of comparing the various formulæ used in designing reinforced-concrete beams, with the behavior of test beams under load. Of the three theories the straight-line theory A is the simplest, and to the writer this still seems the best from a practical standpoint.

2 The formula derived by this theory as now used for determining the depth of a reinforced concrete rectangular beam (using the notation adopted by the Joint Committee on Concrete and Reinforced Concrete[1]) may be expressed simply as

$$d = C\sqrt{\frac{M}{b}}$$

and the ratio of steel required is $A_s = pbd$

where d = depth of beam from compressed surface to center of steel, in inches.
C = a constant for a given steel and a given concrete.
M = moment of resistance or bending moment in general, in inch pounds.
b = breadth of beam, in inches.
A_s = area of cross-section of steel, in square inches.
p = ratio of cross-section of steel to cross-section of beam above the center of gravity of the steel.

3 Theory B, where the stress is taken as varying according to a parabola, is perhaps more exact than theory A, but at the same time more complicated and difficult in practical application. Theory C agrees more closely in the earlier stages of loading with the tests, although tests made both in the United States and in Europe indicate that Considère was not entirely correct in his assumption that steel when combined with concrete permits the concrete to stretch to a greater degree than when not reinforced. However, at earlier stages of loading the cracks in the concrete do not extend up to the neutral axis, so that more or less of the concrete is resisting tension and assists the steel in taking the stress. For this reason a method taking into account the tensile value of concrete gives results closer to the tests at early periods of loading than either formula A or B.

[1]The Joint Committee is composed of representatives from the American Society of Civil Engineers, the American Society for Testing Materials, the American Railway and Maintenance-of-Way Association, and the Association of American Portland Cement Manufacturers.

There are, however, quite important reasons, as will be shown in succeeding paragraphs, why theory A is preferable.

4 Reinforced concrete is a complex material, which if properly used gives very safe and satisfactory structures. It is not, however, of a kind to which hair-splitting accuracy may be applied. In selecting a formula to use, the aim should be to choose one which will give results always on the safe side and at the same time not very wide of the mark. Referring to the paper, formula A gives results on the safe side, while C errs nearly as often on one side as on the other.

5 The behavior of a reinforced-concrete beam under load may be divided into two stages, the earlier stage where the concrete under the neutral axis bears tension, which gradually merges into the later stage, when the tensile strength of concrete is overcome and all the tensile stress is taken up by the steel. In the earlier stage the stress in steel increases proportionally to the moment, while in the later stage the increase in stress in steel is composed not only of the increase proportional to the moment, but also of the stress which in the previous stage was carried by the concrete and after its cracking transferred to the steel. Thus, for example, if a certain load W stresses the steel up to, say 16,000 lb. per sq. in., an addition to the load of *less* than W will double the stress. Therefore, a beam designed for a load which would produce an actual stress in steel of 16,000 lb. per sq. in. would have a factor of safety smaller than the ratio of that stress to the elastic limit of the steel. It is safer, then, to base the design on the results at the breaking load rather than on the results at earlier stages of loading, and to use theory A, which at the breaking load corresponds closely to the tests, and so be sure of the required factor of safety. In designing, working stresses and working moments should be used in the formulae.

6 The strongest argument against computing the concrete to bear tension, in practical design, is the fact that reinforced-concrete floors and other structures usually have to be built with joints between two days' work. The bond of the concrete on the joints is imperfect, and consequently the tensile strength of concrete at that point is small and cannot safely be counted upon in design.

7 Theory A is very simple and clear. It has been adopted quite generally in Germany and England, and I believe also in France, although that is the home of Considère, while the Joint Committee in this country has recently adopted it.

8 Theory A when used in figuring deflection does not give very satisfactory results, but this is not an important factor in reinforced-

concrete design. When necessary to compute deflection, a more complicated formula may be used which considers the tensile strength of concrete. The best of such formulæ known to the writer are those derived by Professor Thullie of Austria, which are based on more logical assumptions than are the formulæ of Considère.

9 It must not be forgotten that the computation of the stress in the middle of a supported beam is only one part of the theory of reinforced-concrete design. Just as important as the design of the beam in the center, since reinforced concrete is usually built continuous over several supports, is the design of the ends of the beam, and of no less importance is the part of the design to resist the tendency of the diagonal tension to produce diagonal cracks.

10 It may be said then in conclusion, that although not corresponding strictly with tests, the ordinary straight-line theory is the one which will probably be used for some time to come because of its simplicity, and because reinforced-concrete beams, designed according to this theory, with due regard to other details, will produce with good workmanship, structures which are unquestionably safe and conservative.

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[1] Massachusetts Institute of Technology.

the other formulæ, and hope that the present data may be extended by further tests and computations. As far as actual use in design is concerned, any one of these theories may be safely used, provided a liberal factor of safety is employed, but further study and investigation along the lines indicated may enable us to determine more precisely what the factor of safety should be.

Henry F. Bryant.[1] I would like Professor Lanza to tell in what way the tested beams failed; whether there were distinct signs of failure at the yield point of the steel, and whether that is the definite point of failure in the beams which he tested. I would also like to ask whether as a result of the tests, he has any evidence that exceeding the yield point of the steel, if it is reached without diagonal crack, is the cause of the failure of the beam.

J. R. Worcester.[2] The careful study which the authors have devoted to these eleven beams is of great value, and their deductions show how much can be learned from a few experiments made with care and recorded with scientific accuracy.

2 It seems to the writer, however, that a few other points of interest in the tables are worthy of comment; as, for instance, the fact that in two of the beams, A-1 and A-2, alike so far as dimensions and amount of reinforcement are concerned, there appears to be a variation of 0.1 in. (1.9 per cent) in the actual location of the neutral axis; of 76 lb. per sq. in. (12 per cent) in the stress in the concrete; of 297 lb. per sq. in. (3.9 per cent) in the stress in the steel, and of 0.007 in. (10 per cent) in the deflection.

3 Another remarkable variation in the behavior of beams apparently alike is that of No. 35 and No. 45, where the latter with 80 per cent of the load of the former had the same actual deformations in steel and concrete, indicating the same location of neutral axis, and at the same time 50 per cent greater deflection. These great differences may perhaps be due to the fact that No. 45 was cracked before the test began, and therefore possibly should be excluded from such a comparison as this, though the cracking did not prevent the beam from developing fairly satisfactory strength. These striking instances of variation in observed results, where every precaution was taken to make the conditions identical, render it important to select theories of computation safe for the worst results found experimentally.

[1]Henry F. Bryant, Boston and Brookline, Mass.

[2]J. R. Worcester, 79 Milk St., Boston, Mass.

4 Speaking from a practical standpoint, several of the elements compared are not of vital importance. The location of the neutral axis is used only as an intermediate step in the process of calculation, and, if fairly correct results can still be obtained, error in this part of the calculation is not serious.

5 Then, again, the deflection is rarely of great importance. It is comforting to know that beams do not deflect as much as if the concrete had no tensile strength, but practically this is as far as we are usually concerned.

6 The actual compressive stress in the concrete may also be eliminated from consideration in actual construction, if only we can limit the area of steel to such a percentage that we are sure failure from the compression of the concrete will not occur until the steel has been stretched beyond the elastic limit. In this connection it is worthy of note that the beams quoted were with one exception more heavily reinforced than is usual at the present time. With 0.8 per cent of steel, or even with 1 per cent, it is safe to base our calculations for moment upon the stress in the steel only.

7 The element then about which the most interest centers is the stress in the steel, and it is important that we should adopt a method of computation which gives this with the least error practicable, and with that on the safe side.

8 Looking at Table 5 with these considerations in mind, we find little difference between methods *A* and *B*, both giving results well on the safe side. Method *C*, while averaging very closely to actual results; gives errors on the wrong side in five out of the eleven cases cited, in one case, and that the one most resembling usual practice, having an error of nearly 15 per cent on the unsafe side.

9 It is noticeable also that the loads assumed are considerably less than what would usually be considered working loads for the beams in question. Following almost universal practice at the present time, the stress in the steel as computed would be allowed to go to 16,000 lb. per sq. in. This would permit loads on the University of Illinois beams as follows:

No. 11, 5,000 lb. in place of 4000 lb.
No. 27, 12,000 lb. in place of 9000 lb.
No. 28, 10,000 lb. in place of 5000 lb.
No. 33, 7,000 lb. in place of 5000 lb.
No. 35, 8,000 lb. in place of 5000 lb.
No. 45, 8,000 lb. in place of 4000 lb.

Only these six are quoted because the essential facts regarding

concrete design. When necessary to compute deflection, a more complicated formula may be used which considers the tensile strength of concrete. The best of such formulæ known to the writer are those derived by Professor Thullie of Austria, which are based on more logical assumptions than are the formulæ of Considère.

9 It must not be forgotten that the computation of the stress in the middle of a supported beam is only one part of the theory of reinforced-concrete design. Just as important as the design of the beam in the center, since reinforced concrete is usually built continuous over several supports, is the design of the ends of the beam, and of no less importance is the part of the design to resist the tendency of the diagonal tension to produce diagonal cracks.

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them are given in the bulletins of the University of Illinois, while we have not at hand the details of the tests at the Massachusetts Institute of Technology.

10 The diagrams of these beams indicate under the above loads the stresses in the steel indicated herewith, using the authors'

STEEL STRESS UNDER HEAVIER LOADING

Beam No.	Load Used	Stress in Steel, Lb. per Sq. In.			Error of Calculation per Cent	
		Actual	By A	By C	By A	By C
11	5,000	15,600	17,700	13,600	+13.5	−12.9
27	12,000	13,500	14,900	12,600	+10.4	− 6.7
28	10,000	14,700	16,600	15,000	+12.9	+ 2.0
33	7,000	12,600	15,200	12,900	+20.6	+ 2.4
35	8,000	13,800	15,750	13,900	+14.1	+ 0.7
45	8,000	15,000	15,750	13,900	+ 5.0	− 7.3
				Average error	+12.75	− 3.6

modulus of elasticity, 30,000,000 lb. In the same table are given the stresses in the steel as calculated by methods A and C, and the percentage of error by each method.

11 Comparing these results with those obtained by the authors as shown in Table 5, we find that the common method of computation A gives considerably closer results to those observed than under the lower loading. The error ranges from 5 to 20.6 per cent, with an average of $12\frac{3}{4}$ per cent, always on the safe side. On the other hand, by the Considère method, C varies from + 2.4 per cent to − 12.9 per cent, with an average of 3.6 per cent on the unsafe side. This would indicate that there is no advantage in adopting the more laborious method, involving the solution of an equation of the fourth degree, at least so far as proportioning the steel is concerned.

12 The chief difference between the two methods, as explained in the paper, is in the assumption in the Considère method of a certain value for tension in the concrete below the neutral axis, and the disregard of this in method A. There is no question that under ordinary conditions the concrete has a small amount of tensile strength while the loads are small, but there is grave doubt as to the safety of relying upon a crystalline material under such conditions. Many conditions in actual construction may tend to destroy the tensile strength.

There may be set-joints near the center of the beam; there may be voids near the bottom where the mortar has leaked out; there may be incipient invisible cracks extending to an unknown distance. It is a fortunate circumstance that ease of calculation is on the side of the safer method, for this is a powerful incentive to its adoption.

13 The statement at the opening and close of the paper that "the observations made thus far are not sufficient to furnish means for determining the actual distribution of the stresses," etc., is undoubtedly true, speaking literally and with scientific accuracy. At the same time it should be borne in mind that we are dealing with a crude product which cannot in practice be made with scientific accuracy. It is doubtful whether absolute knowledge of the laws of distribution of stress in a theoretically perfect material would be of any great advantage in designing structures of every-day material. The important question is whether we know enough to design our beams with entire safety and reasonable economy. To this query the writer would unhesitatingly give an affirmative answer. The investigation of these beams tends to confirm this opinion; which is also supported by the constantly accumulating experience with actual construction. We would therefore venture to add two other conclusions to those advanced by the author, namely:

a Experiments indicate that, though precise determination of the laws of stress distribution may be impossible in the present state of our knowledge, sufficiently close approximations may be made for all practical purposes.

b The simple method of calculation, by neglecting tension in the concrete and assuming a straight-line distribution of the compressive stress, is the easiest to apply and gives satisfactory results for the determination of the stress in the steel.

Prof. Geo. F. Swain. I notice that Professor Lanza has used a value of $E = 2{,}335{,}000$ for the beams tested at the Massachusetts Institute of Technology, while for the beams tested at the University of Illinois he has used a value for E of 2,000,000. The beams tested at the Massachusetts Institute of Technology were from 35 to 54 days old, while the beams tested at the University of Illinois were from 60 to 65 days old. The modulus of elasticity ought to increase with age, other things being equal, yet in these tables the reverse is assumed. This fact might account for some of the peculiarities

and the results. Professor Lanza does not state whether he measured the modulus of elasticity.

2 In Table 2, I think the heading of the column "Nearest one-third Load," is a little confusing. Those figures are not very close to one-third the load, and beam *C*-5, which has a larger load than the first three beams, has a smaller value in the third column. I suppose the third column simply means the loads for which computations were made, and that the loads were applied in such increments that the figures given represent the nearest third of the load for which computations were made. Yet it seems rather confusing that for a load of 16,240 lb., the nearest one-third should be given as 4600 lb.

3 With reference to the three theories, I have never believed in Considère's main contention, namely, that by reinforcing concrete such great strains could be produced without fracture; though his explanation is in a certain degree plausible. If a body is stretched so that the molecules are a certain distance apart, nothing can prevent fracture. Ductile material like steel draws down at the point of fracture and is stretched much more there than on the average through the length of the piece. If concrete were a ductile material, its adhesion to the steel bars might prevent any such phenomena as drawing down and thus distribute the strain; but concrete is not a ductile material, and there seems to my mind to be no possibility of the great stretch without fracture which Considère claims.

4 As to the results obtained by the three formulae, I think those given in the tables were precisely what might be expected, because these loads were only large enough to be called working loads; that is, they were nothing like the ultimate load. As a matter of fact there was tension in the concrete, under which condition the steel would be relieved; we would therefore expect that in case *C* the stress in the steel would be very much less than in the other two cases. In practice, also, there is undoubtedly tension in the concrete unless cracks occur. The results of tests made by the Boston Transit Commission show large tensile stresses in concrete beams without reinforcement.

5 However, the question is, what to do in designing. In practice there may be cracks in the concrete, not due to stress, but to the moving of blocks on which the rods are set, making the cement run out, or due to shrinkage or joints or other causes; for which reason it seems to me that in practical designing, engineers are not justified in assuming any tension in the concrete.

6 If Professor Lanza tested the modulus of elasticity of his beams,

I would like to know what was the variation in the moduli of elasticity. Was 2,335,000 the averag:? How did the separate values compare?

Henry F. Bryant.[1] Mr. Worcester stated (Par. 9) that on applying his reasoning to the University of Illinois experiments, the nearest one-third load for 16,000 lb. of stress on the steel would be found to be nearly double that given in the paper as approximately one-third the breaking load. This emphasizes the question of the yield point. The rather common practice, as Mr. Worcester states, is to take from 12,000 lb. to 16,000 lb. on mild steel and with this to use about 500 lb. as the concrete compressive strength, which, with concrete of 2000 lb. compressive strength, gives a factor of safety of four or possibly five. If the yield point is the critical point in the steel, we are using a factor of safety of only between two and three in the steel. Mr. Worcester's analysis of the Illinois experiments would indicate that instead of breaking at three times what would be considered a safe working load, the beam would break at not over twice the load. I think that using mild steel and a factor of at least four, and figuring that the yield point is the critical point of the steel, we should apply to the steel something like 7500 lb. or 8000 lb., with 500 lb. compression on the concrete. That means a little larger percentage of steel than is common practice, though it is not unusual to adopt this reasoning with high-carbon steel. I am very glad to see that these experiments point that way.

Rolf R. Newman.[2] Several technical journals have commented on Professor Marburg's tests of Bethlehem steel, in which he obtains some very low values, saying that they considered a well made concrete beam safer than a large Bethlehem beam, because its composition is more definitely known. I would like to ask Professor Lanza whether in his opinion it is correct to say that as much experimenting is needed upon large Bethlehem beams as upon reinforced-concrete beams.

H. E. Sawtell.[3] Considère's theory of stress distribution agrees very well with the actual tests at about working loads on the eleven beams mentioned in the paper. We know, however, that his theory

[1]Engineer, 334 Washington St., Boston and Brookline, Mass.

[2]Civil Engineer, Boston, Mass.

[3]Structural engineer, with Chas. T. Main, Boston, Mass.

will not agree with breaking-load results as well as either the straight-line or the parabolic theory, which consider that concrete takes no tension stress. We should adopt a theory which will agree quite closely with tests at breaking loads, but which will always be on the safe side for intermediate loads. We can then get a real factor of safety.

2 Referring to Par. 24, it seems likely that when applied to floor beams, a formula will remain only a sort of working hypothesis if our theories are to be based upon test beams which are not more like the beams used in actual practice, and if our compressive value for concrete is based upon plain concrete. The present uncertainty may appear to favor the side of safety, but on the other hand, when too

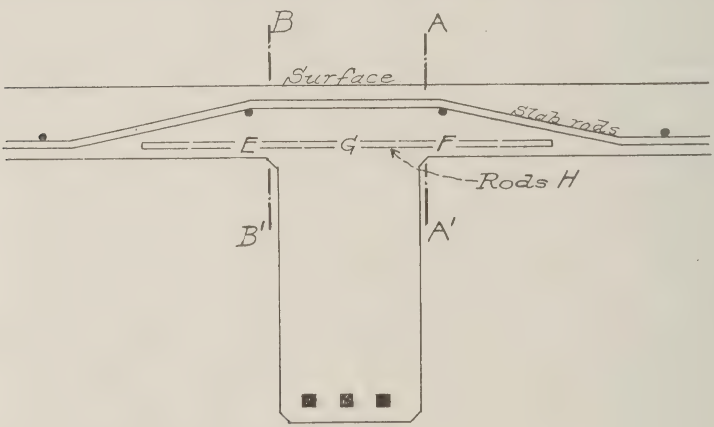

Fig. 1

many assumptions have to be made, there is little real satisfaction in working with the material.

3 Tests on rectangular beams are necessary for determining as nearly as possible the stresses and deflections in slabs and separately moulded beams, but do not seem to solve the problems of beams and girders as used in actual construction. Let us first note some of the stresses as they exist in a beam in actual construction, assuming Fig. 1 to be the cross section of a beam at its place of maximum flexural stress. The slab steel is placed at the beam, as a great many designers consider necessary, in order to resist fully and reliably the negative slab stress, etc., at the beam. These slab rods always are only a few inches apart, and pass through the top of the beam concrete at right angles to the compressive stress of the beam.

4 Assuming that the concrete in both beam and slab is poured at the same time, we know of course that for some distance each way from the beam the slab will work with the beam in resisting compressive stresses. Assumptions are made as to what part of the slabs will work safely with the beam, and then the beam is calculated for and designed as a T-beam. In doing this the full working stress for concrete in compression is used. The concrete at G, E and F has a large share of the compression to take care of. Also, as a result of placing the slab steel at the top, as it passes over the beam, the concrete at G, E and F is again put in compression, this time at its full working value, but at right angles to the compressive stress in the beam.

5 Again, the maximum vertical shear in the slabs is along the lines B-B' and A-A', this shear, it will be noted, being through concrete already doing double duty in compression. The concrete at the surface is at the place of maximum compressive stress of the beam and it also has a maximum tensile stress due to the negative slab moment.

6 The total compression at G, E or F is very much higher than we would willingly put upon plain concrete as a working stress, while the concrete at points E or F is in a worse condition. At the surface the material is nearly cracking from a tensile stress, even under working loads, and it cannot be of much service in compression where it is most needed by the beam.

7 If these conditions are correctly noted, and if the actual stresses are to be kept down to the unit working stress of *plain* concrete, then it will be necessary either to assume a much lower unit stress for concrete when designing T-beams, or to design a rectangular beam whose effective top surface does not extend above the slab rods shown in Fig. 1. But is it necessary to use the value of plain concrete when designing T-beams? Are we not justified in saying that concrete at G is confined, and being reinforced, has a much higher ultimate strength than plain concrete?

8 The compressive strength of concrete in beams is increased in two ways (a) by lateral restraint, brought about by the surrounding compressive forces; (b) by reinforcing its shearing resistance, which may be greatly assisted by placing the rods H at the points shown in Fig. 1. These H rods are to be used only at and near the place of maximum moment in the beam and should be quite close together.

9 But how much does this increase the strength? As bearing upon the subject, an extreme case may be cited from a paper by Leon S. Moisseiff read before the American Society for Testing Materials.

The compressive strength of cubes of concrete, reinforced in every direction by a large percentage of metal in the form of nails, was increased to two to three times the strength of plain concrete. Some designers have already noticed an increase of strength under similar conditions and are taking advantage of it, but are making assumptions regarding its amount for different percentages of reinforcement.

10 So far as the writer knows, no T-beams have been tested with their flanges reinforced and loaded in such a way as to carry their loads to the beam and thus to strain the beam in the same manner as in actual practice. It seems that future tests should be along such a line, if greater economy is to be reached in design and our knowledge is to become more exact with fewer assumptions made.

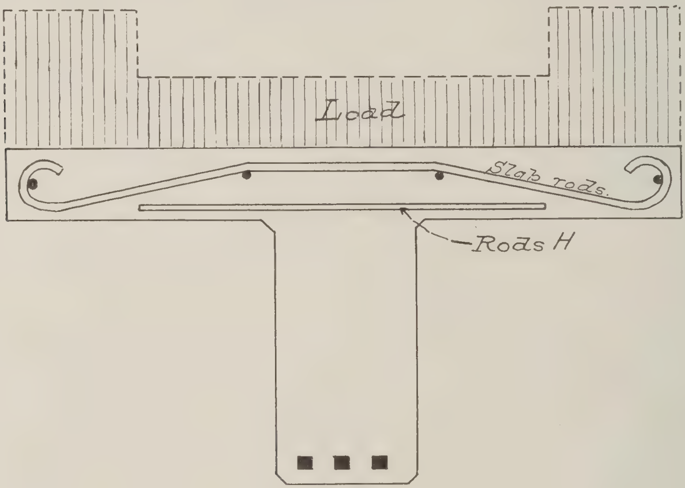

Fig. 2

11 In conclusion, it would seem as though the slab concrete were overstrained at E and F, where it is used for T-flanges, for negative slab compression and for vertical shear from slab loads. Unless it can be ascertained whether lateral restraint, and the use of the rods as shown, will increase the strength necessary to resist this strain safely, it would be better not to calculate for T-beams, but to make the rectangular section sufficient to meet the stress. Even this rectangular section should be designed with a conservative concrete compressive stress, because its top surface is generally considered at about the point where the slab rods pass over it, this including the concrete at G.

12 Fig. 2 shows the cross section at the center of a T-beam, and a method of loading which seems to give promise of results which will come nearer to showing how beams in actual construction are stressed than rectangular beams whose compressive side is wholly plain concrete. The load over the stem should be less than the flange loads; and should agree with actual floor loading where the slabs carry most of the loads to the beam and produce tension in the rods and concrete at the surface over the stem, compression at the under side of the slab at the stem and shear near the stem. As tie rods are always used in practice it would be well to use them here. They are shown by dots in the diagram. The slab rods in this case are bent to act as anchors, and the tie rod at the edge is wired to them on the inside.

13 It is acknowledged that the loads on the flanges do not stress them quite as they would be stressed in a floor system; but if the compression, tensile and shear stresses are not more than those that would be produced, were the slabs continuous, it is thought that as their stress is at right angles to the beam this difference will make no practical difference with the results on the beam.

DISCUSSION AT NEW YORK

E. P. Goodrich.[1] The several theories which were the basis of the formulæ used by Professor Lanza are approximations to actual conditions, and are made the basis for calculating special points in construction work. The first method is used primarily because of its ease of application to ordinary conditions, and the factors now introduced into the formulæ are based almost exclusively on the results of actual tests. For instance, in the particular series of tests made at the Massachusetts Institute of Technology the ratio of the modulus of elasticity as found by experiment to the computed value is only eight and a fraction. On the other hand, diagrams of Professor Talbot's beam tests, in which the position of the neutral axis is shown, give a ratio of more nearly eighteen, showing that the factor introduced has no real relation to actual conditions. It is the adaptation of a formula to tests, rather than the use of a formula to check various kinds of investigations. Occasionally the straight-line formula has been used to compute deflections and stiffness, as was reported not long ago in an article published in Engineering News; but as to the accuracy of this use there has been some adverse criticism.

[1] Consulting Engineer, 1 Madison Ave., New York.

2 As has been said, Considère's theory was based on certain experiments, the accuracy of which has also been questioned. Professor Mörsch of Zürich argues both for and against them in his book entitled Eisenbetonbau, describing certain experiments with concrete beams, in which he determined the stress-strain diagram for both tension and compression, finding some such conditions as that shown in Fig. 1. If in any beam section, the neutral axis be established, and the actual stresses laid down graphically above and below this neutral axis at any point, and if the centroids in each section are determined, and the distance between them measured, the moment which must theoretically be sustained by the beam can be computed. Mörsch tested

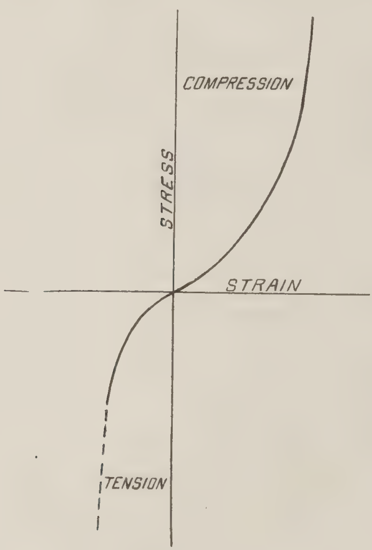

Fig. 1

some specimens both in compression and tension, and also in bending, and computed the theoretical bending moment and ultimate strength by methods similar to Considère's, using a practically constant stress in the concrete below the neutral axis. He found that the theoretical bending stress in kilograms per square centimeter was 20.7, while that found as an average of three actual experiments was 21.4, showing a very close agreement in this particular instance.

3 In the case of three other beams in which the percentage of steel varied from one-half of one per cent to very nearly two per cent, Mörsch made a similar computation based entirely on a stress relation

similar to that of Fig. 1. He found the resultant of the two tensile stresses, in the concrete and the steel, then measured the distance on his diagrams between the centroids of compression and tension, and computed the moment, which was found to correspond closely with the test conditions.

4 Another series of tests of considerable interest is that made by Dr. Mueller for his doctor's thesis for the Hanover Technical High School. He treated concrete beams in a manner similar to that of Professor Lanza, except that he used thirteen points in the depth of the beam, and measured by three methods the actual strain relation which existed at different times. In all his work he used simply a safe working stress, to the limit allowed by the German Government regulations. He found that in a solid beam the stress varied to a certain extent, was very nearly of the straight line type when measured at all his thirteen points; while with a beam in which he built in fourteen artificial cracks by putting sheets of metal close together in the beam, he found that the stress relation more nearly corresponded

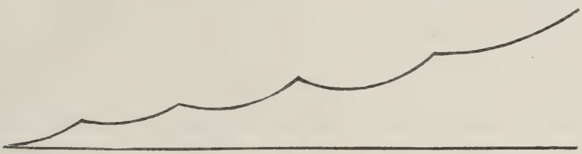

Fig. 2

with Considère's theory These artificial cracks produced a variable stress between the sections, so that the stress in the steel was actually less between the cracks, some of the stress being thrown into the concrete, as illustrated graphically in Fig. 2, in which the ordinates above the base measure the tensile stress.

5 The question of shear has been mentioned, but its effect upon deflections has not been discussed. The writer believes this is very important, because of two series of tests which he made some years ago on beams, one series of which was reinforced only by horizontal rods, and the other by vertical stirrups also. The deflection was three or four times as much in the case of the beams without the vertical steel—shear reinforcement—as in the case of beams with considerable vertical reinforcement. Each series had exactly the same amount of steel in tension. Of course theoretically the vertical stirrups could not affect the tensile stresses in the bottom of the beam. The ordinary theory by which deflection is computed does not include

a factor for shear, which actually does have some effect on the deflection, both theoretically and, as shown by these tests, practically. It must be taken into account, as well as the tension in the concrete, if the actual conditions in the beam, especially with regard to stiffness and deflection, are to be considered.

6 It seems necessary that some relation between deflection and stress should be definitely determined, because deflections can be more easily measured in any beam test than any other phenomena. Almost every novice determines the deflection, although he does not know the relation between it and the stresses involved. It is only through discussions such as this that some true basis can be reached for the computation of the stresses involved in continuous members.

7 There is another point concerning which the writer has made some experiments. By means of plaster of Paris, ordinary sharp carpet tacks were applied to the sides of a beam, with the points sticking outward. The beam was loaded centrally, and the actual deflection curve was simply picked through a piece of paper from time to time as the load was increased. The curves were then enlarged and used as a basis for comparison with the theoretical elastic curve of a beam loaded centrally. There was a very large discrepancy, which was more nearly coördinated when it was assumed that the load was distributed over a length something like one and one-half or two times the height of the beam. It is to be hoped that experiments will be made in regard to the deflection of beams and the distribution of stresses, so that some true relation can be determined, between this element, which is easily measured, and the other elements which are usually unknown: that is, in regard to the relation between deflections and the actual stresses of compression and tension.

Prof. Walter Rautenstrauch. I regret that more observations are not recorded and plotted in the paper and that the methods of making the computations and obtaining the data are not given. It would be interesting to plot the variation of deflection with load as observed, and as computed by the three formulæ selected for comparison.

2 I would ask Professor Lanza how he made his observations for the strain in both concrete and steel and also how he determined from these the neutral axis of the section. If these data were submitted it would be possible to make a comparison with results obtained by assuming other possible values of E, for example, and thus to ascertain to what extent the differences reported might be due to assumed and possible actual values.

3 As concrete construction is for the most part monolithic, and very few beams of the particular kind tested are used, I believe it is of much broader interest to investigate methods of measuring strain and computing stress than formulæ for simple beams. It is a fact, I believe, that all the data reported in this paper as actual stresses in concrete—actual stresses in steel—were obtained, not actually, from direct observations, but rather from relations between stress and strain assumed to exist in the concrete or steel. The same I believe is true in regard to the determination of the neutral axis. If Professor Lanza will tell us what assumptions he made in determining these values we will be in a better position to judge their worth.

4 I need hardly call attention to the fact that the modulus of elasticity for concrete in tension and compression is quite variable. It seems to depend upon the age of the concrete and the intensity of the stress. I believe it would have been of some value to take a slice from the end of these beams and obtain a stress-strain diagram, in order to compute the several values of E and the limits of stress for which each value of E is constant. Otherwise the actual values of the stress are not much more reliable than the values as computed by the formulae, since both are computed from assumed relations.

5 It is interesting to note that Formula B is based on a rational assumption concerning the variations in compressive stresses above the neutral axis. The fact has been well established that the stress varies as the ordinates of a parabola, and not as the ordinates of a straight line. On the other hand, I am inclined to doubt the statement of Considère that the concrete on the tension side can undergo an extension much greater than 0.02 per cent without cracking, when the beam is reinforced, whereas when not reinforced the concrete cracks when the extension is from 0.01 to 0.02 per cent. The mere fact that a reinforcing rod is present does not seem sufficient to change the physical properties of the concrete.

6 I believe Professor Turneaure has shown Considère to have been wrong in this assumption. It is not at all unlikely that Considère removed a piece of concrete in which no cracks had developed. Furthermore, if cracks are allowed to develop on the tension side—and this has frequently been observed in beams under working load—might not this crack gradually extend under repeated loading and seriously impair the safety of the structure?

B. H. Davis.[1] Certain practical considerations may be cited to

[1] Assistant Engineer, D. L., & W. R. R., Hoboken, N. J.

illustrate the difficulties confronting the experimenter seeking a rational solution of the deflection problem. Shrinkage is the worst, or perhaps the most indeterminate factor to be eliminated, since it spoils so many carefully performed experiments, being a large cause of the lack of uniformity so generally noted in experimental data.

2 The shrinkage of a concrete block 8 in. sq. by 2 ft. long has been shown to shorten appreciably a bar ½-in. square embedded in it and accurately measured before and after the setting of the concrete around it. This produces an initial tension in the concrete and an initial compression in the steel. In the case of a beam reinforced in only one plane, as perhaps some of the beams tested may have been, these initial strains may largely account for the lack of uniformity in the results obtained.

3 The shrinkage of concrete in setting, nearly always a variable factor, has almost completely upset the theory of arch-ring deflections when the arch centering is struck. Some settle very considerably upon striking the centering, especially when the arch ring is a monolith from skewback to skewback, while others settle hardly at all when alternate voussiors are made and allowed to set and shrink before the ring is keyed. Shrinkage, it has been proved, almost entirely causes this lack of agreement between the theoretical and the actual deflections when arch centers are struck. It would therefore seem logical to assume that the same cause figures prominently in the deflection phenomena of beams.

4 The shrinkage of a beam of large cross section, acting in opposition to that of a smaller beam, has been known to crack the weaker member from top to bottom, breaking up any dependence that might otherwise have been placed upon the concrete in tension, before the beam had been called upon even to support its own dead load.

5 In designing for a given load by the commonly accepted straight-line formulæ for obtaining stresses in steel and concrete, and using the prescribed unit stresses of the building code, a certain factor of safety results. In other words, an overload of two or three times the load assumed in the design, may be applied, and when removed, the structure should be just as capable of supporting the working load for which it was designed as before the overload was applied.

6 Now, granting the conclusion of the author, in Par. 27, that tension in the concrete materially affects the deflection and strength of beams (between certain limits of load), would it not still seem unwise to take advantage of this tension factor in any design where the assumed load limits might be overstepped at some time, leaving

the beam to serve the remainder of its period of usefulness without the tension factor counted upon in its design?

7 Almost every design is over-stressed sooner or later, occasionally by test load, but more often perhaps, because of the enthusiasm of some shop foreman in showing what his building will stand in the way of abuse. For example, loaded cars of gravel and broken stone, and later a 600-class standard-gage locomotive, were run across a machine and erecting shop floor that was designed for a uniformly distributed load of considerably less than one-half the concentrated moving loads applied, this without any apparent damage to the floor.

8 Settlement, which very often upsets carefully made calculations, causes even more indeterminate stresses in reinforced concrete than in other types of construction, this being due to the continuity and the monolithic character of the material. This fact further emphasizes the necessity for conservatism in working formulæ.

9 Construction joints, put in as they usually are, at points of maximum moment, make any reliance upon the concrete in tension entirely out of the question where such joints occur. It is not generally conceded that construction joints so located do materially weaken a beam except in shear.

10 A beam accidentally cracked entirely through near its middle, while being placed in a testing machine, tested higher than the average of several other beams of the same size and reinforcement, showing that a plane of fracture approximately normal to the center line of a beam had not, in this particular case, unfavorably affected the ultimate strength of a beam equally loaded at its third points.

11 Until more is definitely known concerning the shrinkage of concrete and the many other stresses in reinforced-concrete beams at present indeterminate as a matter of conservatism it would seem better to disregard tension in concrete as a moment-resisting factor.

Chas. B. Grady.[1] Professor Lanza and Mr. Smith have clearly brought out the fact that three of the formulae used for the design of reinforced-concrete beams are approximate with a load of one-third the breaking load. The writer will say a few words in reference to the use of these formulae in the design of beams.

2 Formulae *A* and *B*, which are used by a large number of engineers, do not allow anything for the tension in the concrete and therefore must give for rectangular beams results which are mere

[1] Asst. Mechanical Engineer, New York Edison Co.

approximations up to a point at which the concrete fails to act in tension, but the writer believes that if a comparison had been made at say double the load used, Formulae *A* and *B* would have given better results, and possibly nearer those found by actual test, than Formula *C*, especially for the value of σ_s (stress in steel per square inch).

3 In tests of similar beams made at Cornell University by Messrs. Paulus, Tripp and Davis, the average variation in the values of σ_s (stress in steel per square inch) deduced by formula *A* from those found by experiment was 34 per cent with a load of 4000 lb., and less than one per cent with a load of 8000 lb. The above figures are for five beams having an average breaking strength of 13,200 lb.

4 The errors in values deduced by Formulae *A* and *B* are more liable to be on the side of safety than the errors in values deduced by Formula *C*, and while there is no doubt that Formula *C* will give more accurate results when the stress in the steel is comparatively small, it is the opinion of the speaker that Formula *C*, and other formulae making allowance for the tension in concrete, should be used with caution.

5 It is the practice of many engineers to design reinforced-concrete beams in accordance with certain working stresses and to endeavor so to proportion the beam that it will fail by tension, that is, by either breaking or stressing the steel to a point considerably past its elastic limit, thus making the factor of safety a function of the stress in the steel. In such cases, no matter how much the concrete has helped out the steel under working conditions, when the beam is overloaded the steel must take care of practically the entire tension; and therefore the writer believes that it is wiser not to introduce a value for the tension in the concrete into the formulae used in the design of reinforced-concrete beams.

6 The speaker believes that the formulae for deflection deduced by Professor Lanza and Mr. Smith will be of great value to engineers, and that any one of the three formulae will give results accurate enough for practical purposes in figuring the deflections of T-beams, more of which are used in buildings than rectangular beams.

Frank B. Gilbreth. The most important subject related to reinforced concrete, from the standpoint of the mechanical engineer, is the design of forms, for it is the forms that afford the greatest opportunity for the saving of money, and the consequent reduction of price per cubic foot of new buildings.

2 Beams have been designed and built of rectangular section and over 64 ft. 0 in. long, and have beeen perfectly satisfactory. The most successful building of today as well as of the future must be designed with regard to the economical design and use of forms, and not to the greatest saving in the quantity of steel and concrete used. The forms are the most expensive single item of reinforced-concrete work, and a series of papers and discussions on economics of forms will be of more use to the members than any possible study of the savings that might come from refinements in the design of beams.

3 It is by no means rare to see designs for saving concrete where the value of the concrete saved amounts to much less than the cost of the special or odd-sized forms required.

Prof. Wm. H. Burr. Much has been said about the disagreement of theoretical results with the results of experiments. That is an observation which may be made, I believe, in the case of every material which has ever been used by the engineer; scarcely more so of concrete, either plain or reinforced, than of other material. When a comparison of this kind is made, I think we should bear in mind, first, what theory is used.

2 The so-called common theory of flexure probably is not strictly applicable to any reinforced-concrete beam which has been broken. It is a theory which applies to a beam of very small depth, compared with the length of span. This is not the kind of beam usually found either in plain or reinforced concrete, and usually not even in steel. It is not a matter of surprise, therefore, that such a theory does not give the results found by experiment.

3 It seems to me we shall have to proceed with reinforced-concrete beams precisely as with beams of other material, viz: use a simple working hypothesis for the purpose of securing a formula in which empirical quantities determined by experiment may be used. That is the case with wrought-iron and steel beams, with timber beams, and with all other beams, and it is markedly so, even to a greater extent, with columns.

4 The three theories, A, B and C, may be considered in view of the varying conditions at different stages of stress. It would be difficult to show from any results of tests of concrete, that the law of distribution of stress in theory B is justified. There are some tests which show a graphic relation between the intensities of stress and strain, which approximates a parabolic curve, but probably no nearer

than a circular curve or some other. The majority of tests show that line much more nearly straight than parabolic within the limits of stress found in ordinary concrete beams.

5 It is true that concrete has considerable tensile resistance, when it possesses any, but I think there are few engineers who have used much plain or reinforced concrete, who would be willing to trust the tensile part of the beam to carry load, and to be so recognized in the working formula.

6 The result of the slight contraction of concrete, possibly not within the first two months, perhaps not within the first year of its life, is to create fine hair cracks. We do not know how far these enter the mass; they may be only skin-deep, but in some cases they are much deeper. Hence if the beam should show a continuous concrete structure on the tension side for the first two or three months, it does not follow that it is going to remain so. If we are to recognize such a possibility, and it seems to me we would not be justified in neglecting it, the only safe procedure is that usually followed, of neglecting tension in concrete. That does not mean that concrete may not sometimes have considerable tensile resistance. It simply means that such resistance cannot safely be recognized in ordinary concrete work.

7 These cracks may be very much reduced by continual wetting of concrete after it has been put in place. That is one direction in which the concrete work may be improved. We do not wet the concrete nearly enough after the forms are taken away. If it were feasible, concrete should be kept thoroughly wet from three to six months after being put in place. This is not practicable; but after the forms are taken away, the concrete should be kept soaked with water just as long as possible. The contraction will be less and there will be fewer hair cracks, but it will be impossible to eliminate them entirely.

8 We should be sensible, as engineers, in connection with reinforced-concrete work, precisely as we are or ought to be in everything else, and use the simplest possible formula, i. e., the straight-line formula, and not strain after some ultra-refinement which, when we come to examine it, has little or no solid basis. We should resort to proper theories and select a simple working hypothesis, and then use the test beams to determine such empirical coefficients or quantities as will make the resulting formulæ represent actual results as nearly as possible.

Prof. J. C. Ostrup.[1] Within a short time, from fifteen to twenty years, at most, reinforced concrete has gained an enviable position in the construction world, and unquestionably, in spite of many inherent shortcomings, will better its reputation in the future among both engineers and laymen. It is, therefore, to be regretted that the trend of the authors' paper is toward a negative rather than a positive support.

2 It is a well-known fact that the greater number of the deductions and working formulæ obtained from the science of applied mechanics are based upon certain assumptions which to a greater or less, usually less, extent circumscribe the use of such formulæ. The errors resulting from these fundamental assumptions vary considerably with the different engineering materials with which we deal; they often vary considerably even with the same material, changing somewhat with the extreme fibre stress, the manner of application of the load, etc. The assumptions made in regard to the behavior of structural steel are probably nearer the absolute truth than for any other engineering materials, so near, in fact, that many engineers have come to regard the theory of steel design as following an unassailable mechanical law. Nevertheless this is not so.

3 On the other hand, the theory of reinforced concrete is based upon many assumptions, some of which can be better defended than others, and some of which have undergone, and will continue to undergo, modifications from time to time. It is also based upon many widely varying experiments which the experimenters themselves have been struggling to reconcile. Some of the most important of these assumptions, together with a brief account of their probable effect, are:

a That the applied forces in bending are perpendicular to the neutral axis.

4 This is incorrect, of course, inasmuch as the neutral axis under deflection follows a curve resembling a parabola. The resulting error is, however, extremely small.

b That a sectional plane, true before bending, also remains true after.

c That each fibre acts independently of adjacent fibres.

5 The last of these assumptions is particularly faulty, inasmuch as the ordinary reinforced beam usually has its reinforcement vary-

[1] Professor Structural Engineering, Stevens Institute of Technology.

ing in amount, both horizontally and vertically, throughout its length. In other words, unlike a rolled steel beam whose moment of resistance is uniform from end to end, the reinforced beam is not uniform in strength, the stronger parts tending to assist or restrain the weaker. The error from this assumption cannot be evaluated.

d That the concrete and the reinforcement will stretch or compress together without breaking the contact bond between them.

6 This condition, when complied with, as it infallibly must be in all cases, unquestionably sets up secondary local stresses, the magnitude of which cannot be even guessed.

e That there are no initial stresses.
f That the stress-strain curve for compression is a parabola.

7 The fulfillment, or the non-fulfillment, of the last two assumptions, is probably what causes the greatest divergence between theory and tests. A concrete beam is a casting, in a sense. If the mixture were perfectly uniform throughout, there would most likely not be any initial stresses due to the chemical action of setting. This is evidently not possible; hence throughout the beam there undoubtedly exist initial stresses of uncertain magnitude. This fact, in itself, would surely affect the stress-strain curve, but in addition we must consider the variable modulus of elasticity for the concrete. This varies not only in the same beam, according to unit stress in the extreme fibres, but also in beams of the same identical composition according to its depth, i. e., to the relation between the extreme fibre stress and the average fibre stress.

8 In addition to these mechanical considerations, we have many physical considerations governing the strength of concrete and reinforced-concrete beams. Such physical conditions must largely depend upon the personal equation of the engineer in charge; they may be guarded against, and their effect minimized but not wholly eradicated. When present, their influence can only be surmised.

9 To make this a little clearer let us assume a case where a number of beams were to be prepared for a testing machine and where great uniformity naturally would be sought; to insure which, only one grade of cement, one of sand and one of broken stone, would be employed. Next let us look into some of the more important points affecting the strength of concrete, as follows:

a Condition of the cement; whether all the bags in a cargo are of the same age, or manufacturing batch; quantity of carbonic acid contained; degree of moisture (since the outside bags in a stack, and even the outside layer in the same bag, often absorb considerably more moisture than the inside).

b Uniformity of quality of the sand; whether or not it contains in spots, loam, clay or other impurities, etc.

c Uniformity of the broken stone; whether or not the stones are alike in strength and texture; whether or not they are broken to a uniform size, etc.

d Quality or purity of the water; method of mixing the concrete, or difference in methods of mixing from batch to batch, even by the same gang.

e Tamping and placing of the concrete, including the often unavoidable variations in the degree of flexibility of the support between the ends and the center of the beam while the concrete is being tamped.

f Workmanship. A man is not a machine, consequently the materials mixed and the beams made, even by the same gang, will often vary considerably in spite of precautions. May not beams vary much more when made by different sets of workmen?

10 Besides the foregoing points affecting the mechanical laws governing the strength of the concrete, there are others; but enough have been indicated here to show that, when tested, a variation in their strength must exist.

11 Since each experimenter must base his deductions upon the results of his own observations, a divergence in the resulting formulæ is the natural result, and furthermore, were he to repeat the same tests under similar circumstances, his second results, in view of the foregoing, would vary from his first. With all this in mind, is it any wonder that closer agreement between the various working formulæ most generally in use, has not so far been reached? To an unbiased mind the wonder is that the divergences are not even greater.

12 Returning to the conclusions of the author, he states in Par. 24 " . . . . the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulæ . . etc." This may be strictly true in theory, but will hardly be generally accepted as a matter of fact. On the contrary, it is quite within

good reason and good practice to deduce reliable formulæ, even where the action of some of the minor points involved is in doubt, so long as the effective range of such points is known. In this connection it may be recalled that concrete and masonry structures, centuries old, are still standing and doing effective service, though they were designed from formulæ and data far less reliable than those now at our disposal.

13 The author further says: "It follows therefore that whichever of the theories is adopted for practical use, it can be regarded only as a sort of working hypothesis." This, of course, is a sweeping condemnatory statement which, if it can be applied to the theory of reinforced-concrete construction, can, it is believed, be equally well applied to the theories underlying any form of construction; for no amount of theory, unaccompanied by practical experience and sound judgment, will prevail, either in the mechanical or in any other engineering field. This fact cannot be too strongly emphasized.

14 In Par. 26 the author states that theory C gives results in closer agreement with experiments than does either A or B. This is undoubtedly true, but so far as the evidence in Tables 5 and 6 is concerned, any one of the three theories is based upon "reliable formulæ" or, what is more to the point, the designs resulting from their use would be wholly reliable. As a matter of opinion, the preference should be for A or B, since they are nearly correct in regard to the unit stresses in the concrete,—the weaker material,—whereas they give somewhat smaller stresses for the steel than those expected.

15 It is equally true that no reliable deflection formulæ can be deduced without taking into consideration the tension in the concrete. We can, however, go a step further, and state that such formulæ, to be correct, must also include a provision for a deflection increment due to shear.

16 In concluding these remarks, the writer would suggest a caution to such alarmists as are prone to appear from time to time against a useful and excellent building material. No public good can result from arousing the apprehension of either engineer or layman with respect to reinforced concrete, and those of us who have had the opportunity of using it for a number of years cannot help but be impressed with its increasing serviceability and scope.

E. Lee Heidenreich.[1] The tests at the Massachusetts Institute of Technology, as well as those at the University of Illinois, were based upon a concrete mixture of 1 : 3 : 6, while those of Considère

[1] Special Engineer, N. Y. C. & H. R. R., New York City.

are based upon a mixture of 1 : $2\frac{1}{2}$: $2\frac{1}{2}$. I have repeatedly at meetings of the "Joint Committee" urged the desirability of employing stronger mixtures, and mixtures of a "maximum density" rather than a certain proportion; and I believe that with such stronger mixtures Formula C will come still nearer to a correct interpretation of stresses and strains. If so, is it not natural to hope that in our reinforced-concrete building constructions, lesser dimensions of beams and girders, thinner floor slabs, and consequently a reduced item of *dead load* will result, also materially reducing the present disadvantages of heavy columns and foundations?

2 The most wonderful constructions of tanks, reservoirs and bridges in Europe, have resulted from mixtures of 1 : 3 or 1 : 5, properly graded. Why should not our beam tests be based upon such mixtures, notwithstanding the fact that at first glance they may not appear commercially advantageous for building constructions? I wish to place myself again on record as advocating a larger percentage of cement and a mixture representing a maximum density of the ingredients.

Prof. C. E. Houghton. The paper adds to our knowledge of the probable magnitude and sign of the errors due to the use of formulæ deduced from a simpler theory. When the size of a structural member has been calculated by the use of a formula known to give a greater value to the unit stress than actually exists, the designer need not worry about the safety of that member. If in addition the probable magnitude of the error is known, corrections may easily be made where it is considered necessary to reduce the cost or weight of the member.

2 The neglect of tensile resistance in calculations of the strength of reinforced-concrete beams finds a parallel in the common practice for the calculation of the strength of riveted joints. The friction between the plates unquestionably adds to the strength of the joint, yet as far as the writer knows, no theory has been accepted in American practice that considers this friction as acting. This friction, like the tensile resistance of concrete, may vary from zero to a maximum value, and therefore should be neglected, as neither can be depended on for additional strength.

3 All formulæ for the strength of reinforced-concrete beams contain a factor whose value is the ratio of the modulus of elasticity of steel to that of concrete, and any error made in the assumption of that value affects the result in the same proportion. The modulus of elasticity

of steel is practically a constant term, but that for concrete varies through a wide range of values depending to a certain extent on the proportions of cement, sand and broken stone used in the concrete.

4 With the large possible variation of this ratio in mind, it would seem reasonable to suppose that the probable error, either in assuming a straight-line law for the variation of the compressive stress in the concrete, or in the neglect of its tensile resistance, will be less than that due to the choice of the value of this ratio. What is needed is a value for this ratio, determined by applying the formula derived from the straight-line no-tension theory, to the results of a great many tests on specially prepared beams.

5 The number of variable conditions that would affect the results in any such investigation is so great that unless one of our national engineering societies will undertake it there seems to be but little prospect of obtaining anything more than an approximate value based on the results of compressive tests on concrete.

Wm. Wallace Christie. The writer is particularly interested in the applications of reinforced concrete in engineering work, and has had to do with the designing of a great many floors, foundations and other work. He agrees with Professor Burr, and others not prepared to accept or consider a theory of design of concrete-steel beams allowing tension in the concrete, or an increase by reinforcement of the ability of the concrete to resist tension.

2 After concrete work has been erected for a time, hair-cracks, and others more decided, often develop in the beams. An example of this has already been cited: a 70 ft. concrete girder, or longer, with its center, at least, resting on hard pine timbers.

3 With the large factor of safety necessary in the design of concrete-steel beams, one cannot go very far wrong in using any of the three methods mentioned, but the writer prefers a straight-line formula.

4 The paper deals in particular with beams, which in practice are seldom used, except as lintels, or over openings in building walls. The experiments conducted with these beams will not give the results obtainable by the use of T-beams, and the writer doubts whether the test of a single T-beam, made in the test room, will develop the same strength or other features, as a test made on a similar T-beam which is part of a floor system. The beam tested in the laboratory is not joined tightly with the rest of the floor, while in actual construction the iron would necessarily be secured to the other parts of the floor system.

The Authors. The data and the results of observation for the first five beams, which have been asked for, are contained in a paper by G. Lanza, published in the proceedings of The American Society for Testing Materials for 1906.

2 The modulus of elasticity of the concrete was obtained from tests made upon seven 8 in. by 8 in. by 60 in. plain compression pieces of the same age, materials and mixture as the beams. The values of E are as follows:

	2,479,000
	2,223,000
	2,367,000
	2,264,000
	2,670,000
	2,623,000
	2,341,000
Average........................	2,424,000

In our paper we have used 2,335000 in order to permit of the use of $r = 12$.

3 It may be added that the neutral axis was determined for each load from the strain diagrams (which are shown graphically in the paper referred to) at the intersection of the plotted line with the vertical datum line. Numerical details of the strains will be given in appendix A, as they seem to be desired.

4 As reference has been made to evidence tending to discredit Considère's theory regarding the ability of concrete to stretch when reinforced, it may be well to say that it is neither the object of the paper to discuss this question, nor to take sides for or against this theory. The history of the main part of the controversy is as follows:

5 The theory was attacked by Kleinlogel in an article published in Beton u Eisen, Hefts 2 and 4, 1904, in the light of certain tests which he had made. The two tests of Considère on page 1038 of our paper were made as a refutation of Kleinlogel's argument. An account of them may be found in Considère's book on reinforced concrete. A subsequent reply by Kleinlogel, and a reply to this by Considère, are to be found in Beton u Eisen, but no new matter is given.

6 In Beton u Eisen, Heft 11–1905, Professor Ostenfeld gives an account of the results of some computations made by him upon the beams tested by Kleinlogel, and in the light of these he says "Thus

far I regard Kleinlogel's tests as a beautiful though unwilling confirmation of Considère's theory." To this Kleinlogel replies in Beton u Eisen, Heft 1-1906, but this reply contains no new evidence.

7 Fear seems to be expressed by some that pointing out the very considerable discrepancies between the results of computation made by theory *A*, and the results obtained by experiment, is equivalent to a condemnation of all structures where theory *A* was used in the computations. No such condemnation, however, is intended by the authors. They believe, however, that the more we realize the facts in any case, the better prepared are we to use our judgment as engineers, in designing any construction.

8 Most of the arguments advanced in support of the entire sufficiency of theory *A* may be summarized as follows:

a The calculations can be more easily made.

b That the mere fact of neglecting the tension in the concrete results in safety, though practically all admit that the concrete does resist tension in the early stages.

c The use of construction joints, which often take the form of a vertical joint at the middle of the span when work on a given floor extends over a period greater than one day.

9 These matters will be considered in the same order:

a There is no doubt that the calculations are more easily made when theory *A* is used.

b Whichever of the three theories is used, it is not customary to calculate by means of it, the stresses which produce diagonal cracks, and it is a fact that in a very large percentage of the beams that have been tested, the failure has been due to these diagonal cracks. Hence it seems to us that until we have arrived at some means of making calculations to determine these stresses and strains in such a way that the calculated results shall have a fair degree of agreement with the results obtained by experiment, we can hardly claim to have an all-sufficient theory. Moreover, in the case of beam A-1, the only one for which the shear has been figured, it is greater when determined from theory *C* than when obtained from theory *A*, the difference being in one case 57 per cent.

c When a construction joint is introduced, the beam is necessarily weak, and until tests of such beams are made, we cannot claim to know what theory will apply to them.

10 Other considerations which it would seem worth while to discuss are the following:

a The presence of initial stresses due to shrinkage.

b The variation in the value of the compressive modulus of elasticity of concrete.

c The recommendation made by some that the formulæ to be used be based upon loads larger than one-third the breaking load, and by some upon the breaking load.

d The question of so proportioning the reinforcement that the breaking shall be due to the tension in the steel exceeding the elastic limit.

11 Discussing these in order we have:

a The presence of initial stress is of course a great source of uncertainty in reinforced concrete, as well as in cast iron, and hence we should expect irregularities due to this cause, the amounts of which are very difficult to estimate. Whether their influence is still large or not at one-third the breaking load, is a debatable question, though it must be comparatively less at one-third than at smaller loads. On the other hand, with loads greater than one-third the ultimate, the ratio of stress to strain becomes quite variable, and any rational formula becomes inaccurate.

b In the light of the experiments made by different men and in different places, it would seem to the authors that the variations of the modulus of elasticity for compressive stresses in the concrete, not more than one-third the ultimate, would not be very excessive.

c In the case of steel or other beams it is well known that the ordinary formulæ do not apply when the stresses in any of the fibres have passed the elastic limit; hence the difference between modulus of rupture and outside fibre stress at breaking.

d Regarding the question whether theory *A* will agree better with experiment when the percentage of reinforcement is kept so low that the elastic limit in the steel will be exceeded before any fibre of the concrete has to bear a stress equal to its crushing strength, the only evidence in the paper is the following: In one case the percentage of reinforcement was as low as 0.99 per cent, and in three others, 1.25 per cent, and in these three cases the discrepancies of theory *A* are large.

12 In general, it seems to us that thus far not enough systematic work has been done by way of experimenting and calculating in order that we may have more accurate knowledge about a number of matters, among which may be mentioned:

a The actual distribution of stresses not merely in the case of longitudinal reinforcement, but also with diagonal and other reinforcements, and also in T beams.

b A study of the diagonal tension, not only at the neutral axis, but elsewhere.

c A study of the conditions necessary that the breakage may always be due to the reinforcement exceeding the elastic limit, and whether diagonal cracks occur in those cases.

d A study of the effect of construction joints.

13 There only remain for discussion a few additional matters raised by different gentlemen. While it appears from the last column of Mr. Worcester's table that method *C* gives average results on the negative side, it must be remembered that they depend upon the value taken for *t* (the tensile strength of the mixture). This table, as well as Table 5, clearly shows that if a slightly lower value of *t* had been used for these six beams, their average error would have been a positive one, and also smaller than that by using *A*.

14 Replying to the question of Mr. Newman, we do not think the discussion of the Bethlehem beams is sufficiently relevant to the matter of this paper to be taken up here.

APPENDIX A

STRAINS FOR THE M. I T. BEAMS.

The strains were measured at four points in the depth of the beam on each side as described in the paper before the American Society for Testing Materials, already referred to. Columns 1, 3, 4, 2, in the following tables give the strains for these points. Points 3 and 1 were one and five inches, respectively, *above* the center of the beam, while points 4 and 2 were one and five inches, respectively, *below* the center.

BEAM A-1

ONE 1-INCH PLAIN ROD.
INITIAL LOAD 1250 LB.

AGE 53 DAYS.
BREAKING LOAD 15000 LB.

Loads Lb.	Strains 1	3	4	2
2250	0.000023	−0.000008	0.000033	0.000060
3250	0.000064	−0.000009	0.000048	0.000108
4250	0.000107	0.000008	0.000057	0.000195
5250	0.000186	0.000022	0.000071	0.000262
6250	0.000228	0.000009	0.000124	0.000352
8250	0.000345	0.000005	0.000186	0.000569
10250	0.000448	−0.000022	0.000274	0.000795
12250	0.000543	−0.000026	0.000337	0.001017
14250	0.000672	−0.000088	0.000466	0.001279

BEAM A-2 FIRST APPLICATION

ONE 1-INCH TWISTED ROD.
INITIAL LOAD 1250 LB.

AGE 49 DAYS
BREAKING LOAD 16500 LB.

Loads Lb.	Strains 1	3	4	2
2250	0.000044	0.000012	0.000003	0.000033
3250	0.000082	0.000012	0.000027	0.000093
4250	0.000138	−0.000013	0.000077	0.000174
5250	0.000172	0.000016	0.000073	0.000251
6250	0.000216	0.000018	0.000108	0.000358
8250	0.000317	−0.000004	0.000202	0.000595
10250	0.000405	−0.000009	0.000271	0.000835
12250	0.000505	−0.000063	0.000391	0.001039

BEAM B-3

TWO $\frac{3}{4}$ IN. PLAIN RODS.
INITIAL LOAD 1250 LB.

AGE 43 DAYS.
BREAKING LOAD 15950 LB.

Load Lb.	Strains. 1st application 1	3	4	2
2250	0.000073	0.000013	0.000017	0.000081
4500	0.000100	−0.000003	0.000059	0.000175
5250	0.000144	0.000015	0.000060	0.000223
6250	0.000195	0.000002	0.000096	0.000289
8250	0.000398	−0.000020	0.000182	0.000428
10250	0.000519	−0.000066	0.000301	0.000587

BEAM C-5

Four ½ Inch Plain Rods. Age 35 Days
Initial Load 600 Lb. Breaking Load 16240 Lb.

Loads	Strains			
Lb.	1	3	4	2
2600	0.000083	0.000018	0.000026	0.000087
4600	0.000219	−0.000024	0.000133	0.000296
6600	0.000337	−0.000067	0.000239	0.000532
8600	0.000444	−0.000059	0.000297	0.000751
10600	0.000542	−0.000091	0.000406	0.001023
12600	0.000631	−0.000137	0.000525	0.001272
14600	0.000765	−0.000209	0.000653	0.001525

BEAM E-9 FIRST APPLICATION

Two ⅜ In. Twisted Rods Age 54 Days
Initial Load 1250 Lb. Breaking Load 21000 Lb.

Load	Strains			
Lb.	1	3	4	2
2250	0.000037	−0.000012	0.000029	0.000037
4250	0.000107	0.000003	0.000046	0.000134
5250	0.000155	0.000008	0.000060	0.000175
6250	0.000202	0.000004	0.000081	0.000256
8250	0.000275	0.000004	0.000122	0.000402
10250	0.000403	0.000010	0.000161	0.000541
12250	0.000486	0.000003	0.000212	0.000680

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M. E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

American Railway Association. Statistical Bulletin No. 59-A. *Chicago*, 1909.

Association of Licensed Automobile Manufacturers. Bulletin No. 18. July 1906. *New York*, 1906.

Baylor University. Report of the President and Trustees September-November 1909. *Waco.* 1909.

Board of Supervising Engineers, Chicago Traction. First Annual Report. *Chicago*, 1908. Gift of the Board.

Boston Transit Commission. Fifteenth Annual Report. *Boston*, 1909.

Calendar of the Sir William Johnson Manuscripts in the New York State Library. *Albany*, 1909. Gift of New York State Education Department.

Check List of Publications of the University of Wisconsin, 1909. *Madison*, 1909.

Chronological History of the Origin and Development of Steam Navigation. Ed. 2. By G. H. Preble. *Philadelphia*, 1895. Gift of Daniel Arthur.

Commercial Deductions from Comparisons of Gasoline and Alcohol Tests on Internal-Combustion Engines. (Bulletin No. 392, U. S. Geological Survey.) By R. M. Strong, *Washington, Govt.*, 1909.

Comparative Tests of Run-of-Mine and Briquetted Coal on the Torpedo Boat Biddle. (Bulletin No. 403, U. S. Geological Survey.) By W. T. Ray and H. Kreisinger. *Washington, Govt.*, 1909.

Final Hearing of Selden Automobile Cases. June 4, 1909.

Hobart College. Address to the Alumni. *Geneva, N. Y.*, 1909.

Incidental Problems in Gas-Producer Tests. (Bulletin No. 393, U. S. Geological Survey.) By R. H. Fernald and others. *Washington, Govt.*, 1909.

Instructions for Reforesting Land. By C. R. Pettis. *Albany, N. Y.*, 1909.

Life of Robert Fulton. By his friend, C. D. Colden. *New York*, 1817.

Machine Automatique a Tailler sans Gabarit les Engrenages Coniques. By Edmond Dubosc. (Extract from La Revue de Mécanique, May, 1905.) *Paris*, 1905.

Machinery. Vol. 1. 1894–1895. *New York*, 1894–1895. Gift of C. E. Kinne.

Manufacturers' Record's Annual Blue Book of Southern Progress, 1909. *Baltimore, Md.*, 1909. Gift of Manufacturers' Record.

The Monist. Complete Index to Vol. 1–22. 1890–1907. *Chicago*, 1908.

New Complete Dictionary of the English and Dutch Languages. Two Parts. Ed. 2. By I. M. Calisch. *Tiel*, 1890, 1892.

One Hundred Ton Modern Cyanide Plant. By C. C. Christensen. (In *Mining World*, Nov. 13, 1909.)

PRESENT ASPECT OF ELECTRIC LIGHTING. By H. W. Handcock and A. H. Dykes. Institution of Electrical Engineers, August 1909. Gift of Calvin W. Rice.

PREVENTION OF INDUSTRIAL ACCIDENTS. No. 1--General. *New York*, 1909. Gift of Fidelity & Casualty Co.

REPORT OF THE TESTS OF METALS AND OTHER MATERIALS, 1908. *Washington*, 1909.

SIERRA AND SAN FRANCISCO POWER COMPANY. Stanislaus Power Development. Reprint from Journal of Electricity, Aug. 21, 1909.

SOCIETY OF ENGINEERS OF EASTERN NEW YORK. List of Members, 1908. *Albany, N. Y.*, 1908.

STATUS OF THE ENGINEERING PROFESSION. By G. A. Thomas. *London*, 1909. Gift of Society of Engineers.

SUR LE CHOIX DE L'OBLIQUITÉ DE LA LIGNE D'ENGRENEMENT POUR LES ENGRENAGES A DÉVELOPPANTE CONSIDÉRATIONS THÉORIQUES ET PRATIQUES. By Edmond Dubosc. (Extract from La Revue de Mécanique, 1903.) *Paris*, 1903.

TECHNIQUE DU BALLON. By G. Espitallier. *Paris*, 1907.

TECHNISCHER VEREIN VON PHILADELPHIA. Statuten und Nebengesetze, 1907. *Philadelphia*, 1907.

THÉORIE DES DÉRAILLEMENTS PROFIL DES BANDAGES. By G. Marié. *Paris*, 1909. Gift of Dunod & Pinat.

UNIVERSITY OF WISCONSIN. Bulletin, Engineering Series. Vol. 4. Nos. 2–5. Vol. 5. Nos. 1–5. *Madison, Wis.*, 1908–1909.

UTILIZATION OF FUEL IN LOCOMOTIVE PRACTICE. (Bulletin No. 402, U. S. Geological Survey.) By W. F. M. Goss. *Washington, Govt.*, 1909.

WIRELESS INSTITUTE. *Proceedings*. Vol. 1. No. 4. *New York*, September 1909.

EXCHANGES

L'AÉROPHILE. Seventeenth Year. No. 21-date. *Paris*, 1909–date.

APPLICATIONS OF ELECTRICITY TO PROPULSION OF NAVAL VESSELS. By W. L. R. Emmet. Society of Naval Architects and Marine Engineers. November 1909.

BUILDING AND EQUIPPING THE NON-MAGNETIC AUXILIARY YACHT *Carnegie* with Producer Gas Propelling Equipment. By W. Downey. Society of Naval Architects and Marine Engineers, November 1909.

DEVELOPMENT OF THE GASOLINE POWER BOAT. By E. T. Keyser. Society of Naval Architects and Marine Engineers, November 1909.

ENERGY CHARTS FOR STEAM. Supplement to Power and the Engineer, March 16, 1909.

ENGINEERING DIRECTORY. No. 49. October 1909. *London*, 1909.

MASTER CAR BUILDERS' ASSOCIATION. Report of the Proceedings of the 43d Annual Convention. *Chicago*, 1909.

MATERIAL FOR HANDLING EQUIPMENTS FOR LAKE VESSELS. By R. B. Sheridan. Society of Naval Architects and Marine Engineers, November 1909.

NEW ENGLAND WATER WORKS ASSOCIATION. Index to the Transactions and Journal to December 1903, inclusive. *Boston*.

Strength of Water Tight Bulkheads. By W. Hovgaard. Society of Naval Architects and Marine Engineers, November 1909.

Structural Rules for Ships. By James Donald. Society of Naval Architects and Marine Engineers, November 1909.

System of Mathematical Lines for Ships. By J. N. Warrington. Society of Naval Architects and Marine Engineers, November 1909.

TRADE CATALOGUES

E. W. Bliss Co., *Brooklyn, N. Y.* Steam turbines, direct connected to generators, blowers, pumps. 10 pp.

Celfor Tool Co., *Chicago, Ill.* Drills, reamers, chucks, and grinding machinery. 29 pp.

George N. Cole, *New York, N. Y.* Cross horizontal folding doors, "Canopy" and "Jack Knife" construction. 18 pp.

William Cramp & Sons Ship and Engine Building Co., *Philadelphia, Pa.* Automobile and Motor boat castings, 45 pp.; Parsons' white brass ingots and manganese bronze ingots, 24 pp.; Propellers for motor boats, 19 pp.; Parsons' white brass, 8 pp.; Dimensions and price list of Parsons' manganese bronze rolled sheets and rods, 10 pp.

Darley Engineering Co., *New York, N. Y.* Bulletin No. 4. Suction conveyor for coal and ashes. 16 pp.

Diamond Power Specialty Co., *Detroit, Mich.* Pamphlet No. 5: Economical production of steam. 8 pp.

Diehl Mfg. Co., *Elizabethport, N. J.* Bulletin No. 102 on type G motors and generators for general power purposes, 10 pp.; bulletin No. 151 on types F and FC motors and generators. 16 pp.

Dodge Coal Storage Co., *Philadelphia, Pa.* Handling and storing coal and ore, 112 pp.; Telpherage, an electrically operated system of transporting material, 60 pp.; Coal storage according to the Dodge System, 55 pp.

Flinchbaugh Mfg. Co., *York, Pa.* Catalogue 9B: York gas, producer gas, gasoline, kerosene and alcohol engines, 48 pp.

General Electric Co., *Schenectady, N. Y.* Tungsten automobile electric lamps, 15 pp.; train lighting with G. E. Tungsten and Tantalum lamps, 5 pp.; Bulletin No. 4702: fire boats of New York, Chicago, San Francisco, etc., 16 pp.

Glasgow Iron Co., *Pottstown, Pa.* Iron and steel plates, muck bars, flanged and pressed work. 94 pp.

Industrial Instrument Co., *Foxboro, Mass.* Foxboro Recorder, November 1909. 16 pp.

Jenkins Bros., *New York.* Jenkins '96 packing. 1 p.

R. K. LeBlond Machine Tool Co., *Cincinnati, O.* Cutter and tool grinders. 40 pp.

Lyon Metallic Manufacturing Co., *Aurora, Ill.* Installation of Lyon Steel Factory Equipment in the George N. Pierce Co.'s Plant at Buffalo. 20 pp.

Magnolia Metal Co., *New York, N. Y.* Anti-friction metal, 10 pp.; Metal used as babbitt, 8 pp.; Metal vs. Genuine (tin) babbitt, 24 pp.

Northern Engineering Works, *Detroit, Mich.* Newten Cupola, with a differential, adjustable tuyere system and an all steel air chamber. 12 pp

W. R. Ostrander Co., *New York, N. Y.* Speaking-tube hardware, electric bells and batteries, electric light material, telephone and telegraph instruments, and general electric supplies. 690 pp.

Robbins & Myers Co., *Springfield, O.* Standard motors and generators. 32 pp.

M. Rumely Co., *La Porte, Ind.* Oil pull tractor for plowing, hulling, and threshing. 18 pp.

Sawyer Tool Mfg. Co., *Fitchburg, Mass.* Price list of machinists' tools. 55 pp.

Scully Steel & Iron Co., *Chicago, Ill.* November 1909 stock list of iron and steel supplies 96 pp.

Seneca Falls Mfg. Co., *Seneca Falls, N. Y.* Catalogue 22-B: screw cutting lathes, speed lathes and wood turning lathes and attachments. 36 pp.

Société des Ateliers Dubosc, *Turin, Italy.* Automatic machine for cutting conical gears, without pattern. 10 pp.

M. Steiner & Co., *Dayton, O.* Steiner gas and gasolene engines. 24 pp.

Stewart Heater Co., *Buffalo, N. Y.* Otis tubular feed water heater, oil separator and purifier. 16 pp.

Charles A. Stickney Co., *St. Paul, Minn.* Power pump feed, and gravity feed engines, 32 pp.; Bulletin No. 1137, 16 pp.

Joseph H. Wallace & Co., *New York, N. Y.* Representative industrial plants in the pulp and paper industries, and power plants. 200 pp.

Westinghouse Electric & Mfg. Co., *Pittsburg, Pa.* Circular 1094: Turbo-generator sets, 39 pp.; Circular 1103: Multiple arc lamps, direct current, 11 pp.; Circular 1177: Materials for switchboard panels, 11 pp.; Circular 1181: Portable direct current Ammeters and Voltmeters, 7 pp.

Williamson Submarine Corporation, *Norfolk, Va.* October 1909 Submarine Bulletin. 3 pp.

Wm. H. Wood, *Media, Del. Co., Pa.* Hydraulic machinery, 55 pp.; Loco fire box and tube plates, 16 pp.

UNITED ENGINEERING SOCIETY

Gift of J. McAllister Stevenson, Jr. and Louis T. Stevenson

Baker, T. Treatise on the Mathematical Theory of the Steam Engine. *London, 1864.*

Bennett, F. M. Steam Navy of the United States. *Pittsburgh, 1896.*

Bourne, John. Handbook of the Steam Engine. *New York, 1865.*

Haupt, Herman. General Theory of Bridge Construction. *New York, 1866.*

Nason, H. B. Manual of Qualitative Blowpipe Analysis. *Philadelphia, Pa., 1881.*

Nystrom, J. W. Technological Education and the Construction of Ships and Screw Propellers, for Naval and Marine Engineers. Ed. 2. *Philadelphia, 1866.*

Perry, M. C. United States Japan Exhibition. Vol. 1–3. *Washington, 1856.*

Turnbull, John. Short Treatise on the Compound Engine. *Glasgow, 1873.*

U. S. Coast Survey. Coast Pilot of Alaska. Pt. 1. 1869. *Washington, 1869.*

U. S. Navy Department. Report of the Secretary. 1867, 1873, 1876, 1880, 1885. Vol. 1. 1887. *Washington, 1867, 1873, 1876, 1880, 1885, 1887.*

U. S. Navy Department, Office of Naval Intelligence. Annual, July 1892. *Washington, 1892.*

WARD, J. H. Elementary Instruction in Naval Ordnance and Gunnery. *New York, 1861.*

Gift of Prof. F. W. Hutton

INTERNATIONAL EXPOSITION, St. Louis, 1904. Official Catalogue. Exhibition of the German Empire. *Berlin.*

TRADE CATALOGUES

AMERICAN MACHINE COMPANY, *Louisville, Ky.* Full magnet control electric elevators for passenger and freight service, 14 pp.; Description of ammonia regulator for refrigerating machines, 2 pp.; Description of dehydrator for ice and refrigeration machines, 2 pp.; Absorption system of ice making compared with the compression system, 2 pp.; Catalogue of ice and refrigerating machinery absorption system, 37 pp.

VILTER MANUFACTURING CO., *Milwaukee, Wis.* Catalogue A: Refrigerating and ice making machinery. July 1909; Catalogue F: Ammonia fittings for refrigerating and ice making plants; Partial list of users of improved ice making and refrigerating machinery. April 1909.

COMMENT ON CURRENT BOOKS

Large Gas Engines. By Percy R. Allen. Reprinted from Cassier's Magazine, 1909. Cloth, 7 by $9\frac{1}{2}$; 61 pages; 22 illustrations.

The author has divided his subject into three parts: the four-cycle engine—British and Continental practice; the four-cycle engine—American practice; and two-cycle engines. He has described the characteristics of each type at some length, numerous illustrations showing assembled engines and details of construction.

Cyrus Hall McCormick, His Life and Work. By Herbert N. Casson. *A. C. McClurg & Co., Chicago,* 1909. Cloth, $5\frac{1}{2}$ by 8; xii + 264 pages; illustrated. Price $1.50.

The author is well known to readers of popular periodicals through his serials on The Romance of Steel and The Romance of the Reaper. In the present volume he has told of the early struggles and final success of the man who gave grain culture a wonderful impetus through his development of the reaper.

CONTENTS by chapter headings: The World's Need of a Reaper; The McCormick Home; The Invention of the Reaper; Sixteen Years of Pioneering; The Building of the Reaper Business; The Struggle to Protect Patents; The Evolution of the Reaper; The Conquest of Europe; McCormick as a Manufacturer; Cyrus H. McCormick as a Man; The Reaper and the Nation; The Reaper and the World; Give us this Day our Daily Bread.

Steam Navigation, a Chronological History of its Origin and Development. By George Henry Preble, Rear-Admiral, U. S. N. Second Edition. *L. R. Hamersly & Co., Philadelphia, 1895.* Half morocco, $6\frac{1}{2}$ by $9\frac{1}{2}$; 418 pages.

The author starts with the first recorded steamboat experiment in 1543, at Barcelona, Spain, and continues his narrative up to the year 1882, the time of writing. The matter is arranged chronologically, the dates being placed as side heads, so that reference to the development in any year is easily made. The author has treated his subject in an interesting manner, incorporating something of an anecdotal quality to appeal to the lay reader. The fact that the author spent twenty-five years in collecting his material, speaks for its value as an engineering record.

Morrison's Spring Tables. By Egbert R. Morrison. Published by the author at Sharon, Pa. Cloth, 6 by 9; 84 pages. Price $2.

The author has presented a comprehensive list of formulae and tables for the design of light and heavy helical springs and sheet and plate elliptical springs. The properties of light helical springs have been arranged under graduated values of the fundamental ratio—the ratio of the diameter of the bar (or similar dimen-

sion in other than circular sections) to the mean diameter of the spring. The properties of heavy springs are tabulated under each size of bar or plate. From a table on rectangular and elliptical sections, used in connection with the other tables on helical springs, the properties of such springs may be determined easily by proportion. For helical springs the working basis has been taken as one inch of solid height, and for elliptical springs a plate one inch wide. Calculations are based on a fiber strain of 80,000 lb. per sq. in. The modulus of elasticity is taken as 12,600,000 for helical springs and 25,400,000 for elliptical springs.

CONTENTS: Part I, Formulæ: Notation; Helical, Round Bar, Single Coil, General; Helical, Rectangular Bar, Single Coil, General; Helical, Round Bar, Single Coil, Steel; Helical, Rectangular Bar, Single Coil, Steel; Helical, Concentric Coils; Elliptical, General; Elliptical, Steel. Part II, Mathematical Tables: Fractional Parts of π; Cubes; Fifth Powers. Part III, Spring Tables: Helical Wire, Light Steel Spring Table; Helical, Bar, Machinery and Railroad, Heavy Steel Spring Table, Helical, Rectangular and Elliptical Sections; Elliptical, Sheet, Light Steel Spring Table; Elliptical, Bar Carriage, Medium Weight Steel Spring Table; Elliptical, Plate, Machinery and Railroad Heavy Steel Spring Table; Elliptical, Take-up.

Mechanic's and Machinist's Pocket Book. Edited by Wm. H. Fowler. Second Edition. *Scientific Publishing Co., Manchester, England, 1909.* Cloth, 4 by 6, 448 pages, illustrated. Price 6d.

This information in this book is largely culled from British practice, though the editor has in some cases incorporated data obtained from the United States. This is particularly true of the chapter on gearing, the most extensive section of the book. The chapter on shop practice deals with a variety of subjects such as the tempering and working of metals, pattern making, allowances for fits, and the like. A diary for 1910 forms an appendix to the book.

CONTENTS: Handy References and Tables; Mensuration, Geometry, and Trigonometry; Use of Logarithms and Antilogarithms; Materials Used in Machine Construction; Machine Tool Design; Proportions of Machine Tool Parts; Metal Cutting Tools; High Speed Tool Steels; Drilling and Boring Metal; Screw Threads, Screw Cutting, and Taper Turning; Emery and Emery Wheels; Shop Practice; Wheel Gearing; Belt and Rope Driving, Shafting; Lifting Ropes and Chains.

The Prevention of Industrial Accidents. By Frank E. Law, M.E., and William Newell, A.B., M.E. *Fidelity and Casualty Company of New York, New York.* Paper, 5 by 8; 194 pages; 72 illustrations. Price 25 cents.

The prevention of industrial accidents has been the subject of more than one address, and New York has now a museum exhibiting safety devices for the protection of life and limb, but no literature in book form on the subject has yet appeared, we believe, except the book before us. The information was largely supplied from the company's own experience, but other sources—books, technical journals and trade literature—have also been drawn upon. Those features of boiler, engine and elevator design and operation, which must be carefully considered from the standpoint of preventing accidents, are treated at some length, while the safeguarding of the operatives, in factories in general and those of wood-working machinery in particular, is also considered.

CONTENTS by chapter headings: Introduction; Care on the Part of Employers and Employees; Safety Devices; Steam Boilers; Engines; Electrical Apparatus; Elevators; The Factory; Wood-Working Machinery.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

01 Assistant professorship, in charge of design courses in engines, steam turbines, locomotive or gas engines, with assured advancement to full professorship in few years, for the right man. Institution desirous of having its men do outside work. Want a man of ability and experience. Position would pay initially from $1800 to $2000. Location, New York State.

02 A young technical graduate to carry out a series of brick-testing experiments. Previous experience not necessary. Work to last about one year with opportunity to continue on other work when brick testing is completed.

03 Technical graduate to act as general utility man in testing department of large steel works. Previous experience not necessary.

04 Designer of steam engines, compressors, etc., more especially accurate detailing for economic shop production. Position will pay about $2500. Location, New York.

05 Good opportunity is offered to a man with $15,000 to $25,000 capital, to join in an enterprise with a member who has a wide practical experience in manufacturing an electrical material for which there is an established and increasing demand.

06 Wanted, competent, practical operating engineer as chief engineer refrigerating plant of the Panama Railroad, Cristobal, Isthmus of Panama; experience with both refrigerating and electrical machinery essential. Good pay and quarters furnished. Exceptional opportunity for efficient man.

07 Mechanical Engineer to act as salesman for pipe and boiler covering materials; must be a good mixer without being a spendthrift. Salary $1500 to $1800 to start. Location New York.

MEN AVAILABLE

1 Technical graduate, Member, ten years engineering and sales experience, now employed as sales engineer, desires position in purchasing or sales department. New York.

2 Junior, graduate mechanical engineer, four years' experience design and installation; some experience with small gray-iron foundry.

3 Member, age 34, technical graduate, having experience in machine shop, drafting room, testing, estimating and office. Will consider position as manager of sales, or commercial position requiring a knowledge of machinery or engineering.

4 Graduate Lehigh University, class 1897, twelve years' experience as chief draftsman, designing engineer, mechanical engineer and superintendent. Automatic machinery and particularly that relating to printing and typewriting. Inventive and executive ability. Can handle men and take complete charge of the creation and manufacture of mechanical propositions and especially the development of new projects. Location, vicinity of New York.

5 Experienced designer of sugar machinery, in charge of drawing office, would like engagement with well-known manufacturers, as draftsman or erector; or would accept position as engineer in refinery or on plantation.

6 Member, past ten years chief engineer of complete design and construction of crushing plants, power plants, etc., past eight years entirely given to the design and construction of complete portland cement plants. Can furnish references to satisfy the most critical.

7 Mechanical and structural engineer with experience on furnace and mill design, buildings and general machinery, would like position as engineer or chief draftsman.

8 Mechanical and electrical engineer, at present employed as assistant to general superintendent, desires position as superintendent or engineer with concern manufacturing light or medium-weight machinery, or on engineering contract work. Long experience in both engineering and executive positions.

9 Specialist in steam turbine design, desires to locate with firm building steam or electrical machinery and contemplating the addition to present product of a line of steam turbines.

10 Graduate mechanical and electrical courses, W. P. I., age thirty-one, desires position in engineering or executive capacity. Experienced in engineering-contracting business, and construction; has installed, repaired and operated various types of gas, steam and electrical power equipment. Competent to prepare plans, specifications, estimates and reports. Six years on the Pacific coast and previously in New England. Salary $2500. Location immaterial.

11 Superintendent and manager desires change for larger opportunity; high grade organizer and executive; specialist on equipment, production and costs.

12 Shop manager and mechanical engineer; eighteen years experience in the design, manufacture and installation of heavy steam machinery, including hoisting and blowing engines, compressors, steam and hydraulic turbines. Eleven years in charge of factory operation. Best references.

13 Member desires position as works manager or general superintendent; twenty-four years experience as foreman, superintendent and manager of engineering works manufacturing high-class steam engines, boilers, air compressors and steam pumps; also cement mills. Good organizer and executive. If necessary prepared to invest in the right concern. West or Pacific Coast preferred.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ALEXANDER, Ludwell Brooke (Junior, 1905), V. P., Haggerty Contr. Co., Davidson Ave. and Fordham Rd., and *for mail*, The Hazelhurst, 181st St. and Ft. Washington Ave., New York, N. Y.

APPLETON, Thomas (1893), Supt. of Constr., U. S. Public Bldgs., Alton, Ill.

AUSTIN, Adolph Odell (Junior, 1905), Asst. Engr., Vilter Mfg. Co., Milwaukee, Wis.

BAKER, Charles H. (Junior, 1903), 10 Relay Pl., Stamford, Conn.

BALDWIN, Abram T. (1899; 1902), Life Member; Solvay Process Co., and *for mail*, 689 Jefferson Ave., Detroit, Mich.

BARTH, Carl G. (1898), Cons. Engr., 1937 N. 33d St., Philadelphia, Pa.

BENET, Laurence V. (1892), Administrateur-Directeur, Société Anonyme des Anciens Établissements Hotchkiss & Cie, 21, Rue Royale, and *for mail*, 1, Ave. de Camoens, Paris, France.

BRANDON, Geo. Russell (1897; 1901), Harvey, Ill.

BRUSH, Frederick F. (Junior, 1900), Earlimart, Cal.

COLLETT, S. D. (1902), V. P. and Eastern Mgr., Elev. Supply & Repair Co., 114 Liberty St., New York, and *for mail*, 365 Sterling Pl., Brooklyn, N. Y.

CONRAD, Hugh Vincent (1887; 1891), Westinghouse Air Brake Co., Wilmerding, Pa.

GRIESS, Justin, Jr. (1898; Associate, 1908), Treas. and Sales Mgr., Interstate Engrg. Co., Builders Exchange, O.

HARRIS, Grenville A. (1907), Ch. Engr., Takata & Co., 50 Church St., New York, N. Y., and 176 Stiles St., Elizabeth, N. J.

HARTNESS, R. B. (Associate, 1903), 515 W. 124th St., New York, N. Y.

HEALD, Geo. W. (Junior, 1899), 7546 Eggleston Ave., Chicago, Ill.

HEALY, Frederick E. (1906), Mech. Engr. and Spec. Agt., Alberene Stone Co., and *for mail*, 415 3d St. N. W., Washington, D. C.

HECKER, H. A. (1906), 2032 Elm Ave., Norwood, O.

HUSSEY, Charles W. (Junior, 1908), 33 St. Andrews Pl., Yonkers, N. Y.

HYDE, Chas. E. (1885), 940 Fox St., Bronx, New York, N. Y.

LAVERY, Geo. L. (1886), Pres., Amer. Bank Equipment Co., 1315 Old Colony Bldg., and 4300 Ellis Ave., Chicago, Ill.

McCLATCHEY, A. F. (1889), 132 N. 4th St., Aurora, Ill.

McGEORGE, John (1891), Cleveland Engrg. Co., Cons. Engrs., New England Bldg., Cleveland, O.

MAHL, F. W. (Junior, 1892), Asst. to Dir. Maintenance and Operation, Union Pacific System and Southern Pacific Co., 135 Adams St., Chicago, and *for mail*, 1019 Michigan Ave., Evanston, Ill.

MILNE, James (1907), Cons. Engr., 304 Loo Bldg., Vancouver, B. C.

MONROE, Wm. Stanton (1896; 1901), Mech. Engr., Sargent & Lundy, 1720 Ry. Exchange Bldg., and 1235 N. State St., Chicago, Ill.

MORSE, Everett Fleet (1901), Morse Thermo Gage Co., 208 E. State St.,and 111 Eddy St., Ithaca, N. Y.

NICKLIN, Ernest W. (1900; Associate, 1907), Mech. Engr., Detroit Brass Wks., and *for mail*, 421 Cadillac Blvd., Detroit, Mich.

PERRY, Wm. A. (1880), 1 Nassau St., and *for mail*, 7 E. 56th St., New York, N. Y.

ROWE, George F. (1908), 57 Penobscot St., Bangor, Me.

ROYLE, Vernon Elmer (Junior, 1905), Mech. Engr., John Royle & Sons, and *for mail*, 823 E. 28th St., Paterson, N. J.

SAMPLE, Morris De F. (Junior, 1905), Secy-Treas., Fire Protection Co., and *for mail*, 2901 Washington Blvd., Indianapolis, Ind.

SLEE, Norman S. (Junior, 1909), Engr. and Draftsman, Babcock & Wilcox Co., and *for mail*, 410 W. Park Ave., Barberton, O.

SMITH, Orin G. (Junior, 1899), Platt Iron Wks., 1224 Chemical Bldg., St. Louis, Mo.

SMITH, Otto T. R. (1906), Asst. Engr., Engrg. Dept., Otis Elev. Co., 17 Battery Pl., and *for mail*, 880 St. Nicholas Ave., New York, N. Y.

SORNBERGER, Edwin C. (1890), Allis-Chalmers Co., Ellicott Sq., and *for mail*, 208 Lancaster Ave., Buffalo, N. Y.

SWEET, Franklin (Junior, 1903), 285 Farwell Ave., Milwaukee, Wis.

THOMPSON, Edward P. (1884), M. E., Registered Pat. Atty., 1371 Columbia Rd., Washington, D. C.

WHITE, Edward F. (1891), Cons. Engr., Sulphur Plants, Pres., Rutland Mfg. Co., Rutland, Vt.

WICK, Henry (Associate, 1903), 416 Wick Ave., Youngstown, O.

NEW MEMBERS

AKERLIND, G. A. (1909), Cons. Engr., 664 Monadnock Bldg., Chicago, Ill.

BARKER, Perry (Junior, 1909), Chemical Engr., A. D. Little, Inc., 93 Broad St., Boston, Mass.

BORDE, George U. (1909), Cons. Engr., 914 Hibernia Bldg., New Orleans, La.

BOYER, Frederic Quintard (Junior, 1909), 216 Orchard St., New Haven, Conn.

BROWN, Stephen P. (1909), Engrs.' Club, 32 W. 40th St., New York, N. Y.

BULKELEY, Claude A. (1909), Ch. Engr., Board of Education, St. Louis, Mo.

CHAPMAN, Frank T. (1909), Prop., Chapman Mfg. Co., Marbridge Bldg., New York, N. Y., and Montclair, N. J.

CHESS, Harvey B., Jr. (Junior, 1909), 808 Aiken Ave., Pittsburg, Pa.

CROGHAN, John T. (Associate, 1909), Ch. Engr., Concord Elec. Co., and 15 Capitol St., Concord, N. H.

DAMON, Walter Henry (1909), Supt. of Generating, United Elec. Light Co., 87 Greenwood St., Springfield, Mass.

DILLON, Edward L. (1909), Rep., Fairbanks, Morse Co., and 1330a Clara Ave., St. Louis, Mo.

ERNST, Alfred F. (Junior, 1909), Brighton Mills, and *for mail*, 434 Lafayette Ave., Passaic, N. J.

ESSELSTYN, Horace H. (1909), Engr., Westinghouse, Church, Kerr & Co., 10 Bridge St., New York, N. Y., and *for mail*, 296 Vinewood Ave., Detroit, Mich.

FUCHS, Herman (1909), Mgr., Mexican Dept., Fairbanks, Morse Co., and 3910 Cleveland Ave., St. Louis, Mo.

GILMORE, George F. (1909), Local Engr., Am. Thread Co., and 109 Barre St., Fall River, Mass.

GOETZ, Fred. W. (Associate, 1909), Secy., Goetz & Flodin Mfg. Co., Clybourn Ave. and Willow St., and 5960 Kenmore Ave., Chicago, Ill.

HAZELTON, Robert T. (Junior, 1909), Designer, Bridgeford Mch. Tool Co., 225 Mill St., Rochester, N. Y.

HELLER, H. Howard (1909), Eastern Sales Mgr., Hill Clutch Co., 50 Church St., New York, N. Y.

HENES, Harry Wm. (Junior, 1909), 307 E. Green St., Champaign, Ill.

HOUGHTON, Clyde Arthur (Junior, 1909), P. H. B. & N. C. Ry. Co., Eidenau, Pa.

HUNTER, John (1909), Ch. Engr., Union Elec. Light & Power Co., and 4462 Laclede Ave., St. Louis, Mo.

JONES, William R. (1909), Engr. of Constr., Univ. of Pa., and *for mail*, 550 S. 48th St., Philadelphia, Pa.

KENYON, Wm. Houston (1909), Member of Firm, Kenyon & Kenyon, 49 Wall St., New York, N. Y.

KERR, William C. (1909), Mech. Engr., Philadelphia Rapid Transit Co., 9th and Dauphin Sts., and 3322 N. 17th St., Philadelphia, Pa.

KOCH, George B. (1909), Foreman, Loco. Testing Plant, Pa. R. R., and *for mail*, 809 Chestnut St., Altoona, Pa.

LORD, Chas. Edward (1909), Elec. Pat. Atty., Allis-Chalmers Co., Milwaukee, Wis.

LUNDGAARD, Ivar (Junior, 1909), Industrial Engr., Rochester Ry. & Light Co., and *for mail*, 34 Clinton Ave., Rochester, N. Y.

McCARTHY, Harry (1909), Ch. Draftsman, Natl. Tube Co., and 600 E. Prospect St., Kewanee, Ill.

McMILLAN, Chas. M. (Junior, 1909), Cons. Engr., King Edward Hotel, 145 W. 47th St., New York, N. Y.

MONAGHAN, James F. (1909), Mech. Engr., Waltham Bleachery & Dye Wks., and 2 Oak St., Waltham, Mass.

MORETON, George Wm. (1909), Genl. Supt., Betts Mch. Co., and 1323 Gilpin Ave., Wilmington, Del.

MOYER, Allen V. (Junior, 1909), Asst. Secy., Lyons Boiler Wks., P. O. Box 221, De Pere, Wis.

NEWLIN Alexander Z. (1909), Mech. Engr., Natl. Tube Co., and *for mail*, 600 S. Tremont St., Kewanee, Ill.

NORRIS, William H., Jr., (Junior, 1909), Engr., W. R. Grace & Co., and *for mail*, 1 Hanover Sq., New York, N. Y.

OHMES, Arthur K. (1909), Member of Firm, Nygren, Tenney & Ohmes, 87 Nassau St., New York, N. Y.

PALMER, George W., Jr. (1909), Elec. Engr., Old Colony St. Ry. Co., Boston & Northern St. Ry. Co. and Hyde Park Elec. Light Co., 84 State St., Boston, Mass.

PEDDLE, John Bailey (1909), Prof. Mch. Design, Rose Poly. Inst., and *for mail*, 2117 N. 10th St., Terre Haute, Ind.

RICHARDS, Willard F. (1909), Mech. Supt., Gould Coupler Co., Depew, N. Y.

ROELKER, Carl J. (1909), Cons. Engr., Roelker & Lee, State Bank Bldg., Richmond, Va.

ROHLIG, Georg G. (1909), Genl. Supt., Botany Worsted Mills, and 145 Dayton Ave., Passaic, N. J.

SHERWOOD, Mather Wm. (1909), Genl. Inspr., Board of Aqueduct Commrs., and *for mail*, 1090 St. Nicholas Ave., New York, N. Y.

SMITH, Harry J. (1909), Ch. Engr., Hill Clutch Co., Cleveland, O.

STROTHMAN, Louis E. (1909), Asst. Mgr., Pumping Eng. and Hyd. Turbine Dept., Allis-Chalmers Co., Milwaukee, Wis.

STROUSE, Sidney B. (Junior, 1909), Engr., Pa. Engrg. Co., and *for mail*, 1326 N. Marshall St., Philadelphia, Pa.

STURGIS, Wm. Bayard (Junior, 1909), Asst. Engr., Dover White Marble Co., Wingdale, Duchess Co., N. Y.

TYDEMAN, William A. (Junior, 1909), Secy., Macan Jr. Co., and 108 S. 2d St., Easton, Pa.

VANDERGRIFT, James W. (1909), Supt. National Transit Co., Southern Pipe Line Co., Crescent Pipe Line Co., and Eureka Pipe Line Co., and 665 W. Chestnut St., Lancaster, Pa.

WERST, Chas. Wm. (1909), Genl. Foreman, Erecting Dept., Baldwin Loco. Wks., Philadelphia, and *for mail*, 4603 Greene St., Germantown, Philadelphia, Pa.

PROMOTIONS

MARSHALL, Wm. Crosby (1901; 1909), Asst. Prof., Descriptive Geom. and Drawing, 114 Winchester Hall, S. S. S., Yale Univ., and *for mail*, 201 Edwards St., New Haven, Conn.

SCHREUDER, Andrew M. (1898; 1909), Supt., Phila. Textile Mchy. Co., Hancock and Somerset Sts., Philadelphia, and *for mail*, 6201 Germantown Ave., Philadelphia, Pa.

WALKER, Frederick Wiley (1898; 1909), V. P. and Ch. Engr., Comstock, Haigh, Walker Co., 1018–20 Ford Bldg., Detroit, Mich., and *for mail*, Cedarburg, Ozankee Co., Wis.

DEATHS

METCALF, William. SWINSCOE, Charles. WILLCOX, Chas. Henry.

GAS POWER SECTION

CHANGES OF ADDRESS

CHAPMAN, W. B. (Affiliate, 1908), Pres., Chapman Engrg. Co., 50 Church St., New York, N. Y.

COLLETT, S. D. See mem. Am. Soc. M. E.

HOPKINS, George Jay (Affiliate, 1909), Natl. Ry. Devices Co., 490 Old Colony Bldg., Chicago, Ill.

ROTH, Charles (Affiliate, 1909), Mech. Engr., Liquid Carbonic Co., Chicago, and *for mail*, 220 Marion St., Oak Park, Ill.

NEW MEMBERS

CUMMINGS, Wm. Warren. See mem. Am. Soc. M. E.

CUTLER, Frank G. (Affiliate, 1909), Steam Engr., Tenn. Coal, Iron & R. R. Co., Ensley, Ala.

DAVIS, Harvey N. (Affiliate, 1909), Instr., Harvard Univ., 509 Craigie Hall, Cambridge, Mass.

HAGUE, Charles A. See mem. Am. Soc. M. E.

HOBART, Douglas R. (Affiliate, 1909), Tech. Editor, Collier's, and *for mail*, 65 W. 93d St., New York, N. Y.

JENKINS, Alexander Lewis. See mem. Am. Soc. M. E.

MOSES, Frank D. (Affiliate, 1909), Pres., Gas Engrg. Co., Trenton, N. J.

MOSES, Percival R. (Affiliate, 1909), Cons. Engr., 45 W. 34th St., New York, N. Y.

MYERS, Cornelius T. See mem. Am. Soc. M. E.

SPURLING, O. C. See mem. Am. Soc. M. E.

STEVENS, Henry R. (Affiliate, 1909), Cons. Engr., 610 Bailey Bldg., Seattle, Wash.

STOUT, Oscar M. (Affiliate, 1909), Engr., 972 Dean St., Brooklyn, N. Y.

STRITMATTER, Albert (Affiliate, 1909), Secy. & Treas., Gas Engine Pub. Co., and 224 E. 7th St., Cincinnati, O.

TYLEE, Don O. (Affiliate, 1909), 1233 Washtenaw, Ann Arbor, Mich.

WINSHIP, W. E. See mem. Am. Soc. M. E.

STUDENT SECTIONS

CHANGES OF ADDRESS

CARNAHAN, O. A. (Student, 1909), 212 E. Clark St., Champaign, Ill.
COLEMAN, Wm. F. (Student, 1909), Rm. 337, Association Hall, Champaign, Ill.
HEILMAN, H. C. (Student, 1909), 1005 S. 4th St., Champaign, Ill.
JAPPE, Kurt W. (Student, 1909), Main Belting Co., 1241 Carpenter St., Philadelphia, Pa.
LUND, J. C. (Student, 1909), 305 S. Wright St., Champaign, Ill.
McGINNIS, H. D. (Student, 1909), H. B. Smith Co., Westfield, Mass.
WOLF, J. E. (Student, 1909), address unknown.

NEW MEMBERS

ARMOUR INSTITUTE OF TECHNOLOGY

BAUGHMAN, I. N. (Student, 1909), 3166 Lake Park Ave., Chicago, Ill.
BOLTE, E. E. (Student, 1909), 3757 Ellis Ave., Chicago, Ill.
BYERS, A. A. (Student, 1909), 7321 Union Ave., Chicago, Ill.
CARLSON, H. W. (Student, 1909), 2138 Walnut St., Chicago, Ill.
CROCKER, A. H., Jr. (Student, 1909), 650 Barry Ave., Chicago, Ill.
GENTRY, T. E. (Student, 1909), Hotel Metropole, 23d & Mich. Ave., Chicago, Ill.
GILBERT, J. B. (Student, 1909), 3325 Armour Avenue, Chicago, Ill.
GRENOBLE, H. S. (Student, 1909), 4312 Champlain Ave., Chicago, Ill.
GRIFFITH, F. H. (Student, 1909), 3343 Calumet Ave., Chicago, Ill.
HENWOOD, P. B. (Student, 1909), 300 E. 33d St., Chicago, Ill.
LOHSE, A. W. (Student, 1909), 3346 Dearborn St., Chicago, Ill.
McCAGUE, A. (Student, 1909), 140 No. Franklin Ave., Austin, Ill.
PARSONS, H. N. (Student, 1909), 3334 Armour Ave., Chicago, Ill.
THOMAS, W. E. (Student, 1909), 6500 Ellis Ave., Chicago, Ill.
WERNICK, F. E. (Student, 1909), 3316 Dearborn St., Chicago, Ill.
YOUNG, D. A. (Student, 1909), 3332 Armour Ave., Chicago, Ill.

BROOKLYN POLYTECHNIC INSTITUTE

SMALL, G. S., 3d. (Student, 1909), 61 Pierrepont St., Brooklyn, N. Y.

CORNELL UNIVERSITY

BATT, I. A. (Student, 1909), 115 College Ave., Ithaca, N. Y.
BOWER, F. A. (Student, 1909), 58 Thurston Ave., Ithaca, N. Y.
BROWN, C. S. (Student, 1909), 1 Central Ave., Ithaca, N. Y.

CANADY, M. S. (Student, 1909), 518 Stewart Ave., Ithaca, N. Y.
COMINS, H. N. (Student, 1909), 438 Cascad Bldg., Ithaca, N. Y.
CROSSMAN, D. M. (Student, 1909), 105 De Witt Pl., Ithaca, N. Y.
FAIRBANKS, F. L. (Student, 1909), 422 E. State St., Ithaca, N. Y.
GOLDBERG, M. S. (Student, 1909), 102 Highland Pl., Ithaca, N. Y.
GRAY, F. R. (Student, 1909), 113 De Witt Pl., Ithaca, N. Y.
HARDING, H. G. (Student, 1909), 704 E. Buffalo St., Ithaca, N. Y.
LINDSAY, H. D. (Student, 1909), 415 Stewart Ave., Ithaca, N. Y.
NIXDORFF, S. P. (Student, 1909), 221 Eddy St., Ithaca, N. Y.
PEACH, P. L. (Student, 1909), 306 Eddy St., Ithaca, N. Y.
REINICKER, N. G. (Student, 1909), 203 Williams St., Ithaca, N. Y.
REYNOLDS, H. B. (Student, 1909), 203 Williams St., Ithaca, N. Y.
SERRELL, J. J. (Student, 1909), 102 West Ave., Ithaca, N. Y.
SKINNER, H. A. (Student, 1909), Sheldon Court, Ithaca, N. Y.
TURNER, E. T. (Student, 1909), 404 Stewart Ave., Ithaca, N. Y.
UNCKLES, H. W. (Student, 1909), 226 Eddy St., Ithaca, N. Y.
WALL, R. E. (Student, 1909), 110 Osmun Pl., Ithaca, N. Y.
WESLEY, C. F. (Student, 1909), 203 College Ave., Ithaca, N. Y.
WING, S. R. (Student, 1909), 208 Dryden Rd., Ithaca, N. Y.
WOOD, A. P. (Student, 1909), 130 Dryden Rd., Ithaca, N. Y.
WOOD, S. V. (Student, 1909), 110 Osmun Pl., Ithaca, N. Y.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

PAGE, Atwood C. (Student, 1909), 137 Newbury St., Boston, Mass.

UNIVERSITY OF ILLINOIS

KEOWN, B. L. (Student, 1909), 511 E. White St., Champaign, Ill.
MOSCHEL, H. (Student, 1909), 405 E. Green St., Champaign, Ill.

UNIVERSITY OF KANSAS

BRIGHAM, C. M. (Student, 1909), 23 E. Lee St., Lawrence, Mass.
HILFORD, W. H. (Student, 1909), 1025 Kentucky St., Lawrence, Kansas.
JOHNSON, C. E. (Student, 1909), 736 Maine St., Lawrence, Kansas.
PLANK, Wm. Jay (Student, 1909), 814 Alabama St., Lawrence, Kansas.

COMING MEETINGS

JANUARY AND FEBRUARY

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the Editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

ALBERTA ASSOCIATION OF ARCHITECTS

January, annual meeting, Edmonton. Secy., H. M. Whiddington, Strathcona.

AMERICAN MATHEMATICAL SOCIETY

February 26, New York and San Francisco sections. Secy., F. N. Cole, 501 W. 116th St., New York.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

January 18–20, annual meeting, 29 W. 39th St., New York. Secy., W. M. Mackay, Box 1818.

AMERICAN SOCIETY OF HUNGARIAN ENGINEERS AND ARCHITECTS

January 8, 29 W. 39th St., New York. Paper: Measurement of Feeble High Frequency Currents, Aurel Kozmutza. Secy., Zoltán de Németh.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

January 11, February 8, 29 W. 39th St., New York. January 15, St. Louis. January 21, Boston. May 31–June 3, Spring Meeting, Atlantic City, N. J. July 26–29, joint meeting with Institution of Mechanical Engineers, England. Secy., Calvin W. Rice, 29 W. 39th St.

AMERICAN SOCIETY OF SWEDISH ENGINEERS

January 8, annual meeting, 271 Hicks St., Brooklyn, N. Y. Secy., E. Hammerstrom.

ASSOCIATION OF ONTARIO LAND SURVEYORS

February 22–24, annual meeting. Secy., Killaly Gamble, 703 Temple Bldg., Toronto.

BOSTON SOCIETY OF ARCHITECTS

January 4, annual meeting. Secy., E. J. Lewis, Jr., 9 Park St.

BOSTON SOCIETY OF CIVIL ENGINEERS

January 26, annual meeting, Chipman Hall, Tremont Temple. Secy., S. E. Tinkham, 60 City Hall.

CANADIAN SOCIETY OF CIVIL ENGINEERS

Quebec Branch, January 21, annual meeting, Montreal. Secy., C. H. McLeod, 413 Dorchester St., W.

CIVIL ENGINEERS SOCIETY OF ST. PAUL

January 10, annual meeting. Old State Capitol Bldg., 8 p.m. Secy., D. F. Jurgensen, 116 Winter St.

ELECTRIC CONTRACTORS' ASSOCIATION OF NEW YORK STATE
January 18, Utica, N. Y. Secy., Geo. W. Russell, 500 Fifth Ave., New York.

ENGINEERS CLUB OF PHILADELPHIA
February 5, annual meeting, 1317 Spruce St. Secy., W. P. Taylor.

ENGINEERS SOCIETY OF PENNSYLVANIA
January 4, annual meeting, Harrisburg. Secy., E. R. Dasher, Gilbert Bldg.

ENGINEERS SOCIETY OF WESTERN PENNSYLVANIA
January 18, annual meeting. Secy., E. K. Hiles, 803 Fulton Bldg., Pittsburg.

FRANKLIN INSTITUTE
January 28, February 11, Witherspoon Hall, Philadelphia, Pa. Lectures: Road Administration and Maintenance, L. W. Page; Recent Methods for the Production of Light, R. H. Bradbury.

ILLINOIS SOCIETY OF ENGINEERS AND SURVEYORS
January, annual meeting, Cairo. Secy., F. E. R. Tratman, 1636 Monadnock Blk., Chicago.

ILLUMINATING ENGINEERING SOCIETY
January 11, Royal Society of Arts, John St., Adelphi, London. Paper: Glare, its Causes and Effects, J. H. Parsons. Secy., L. Gaster, 32 Victoria St.

INDIANA ENGINEERING SOCIETY
January 14–16, annual convention, Indianapolis. Secy., Chas. Brossmann, Union Trust Bldg.

IOWA ENGINEERING SOCIETY
Fecruary 16–17, Cedar Rapids, Ia. Secy., A. H. Ford, Iowa City.

LOUISIANA ENGINEERING SOCIETY
January 8, Hibernia Bldg., New Orleans, La. Secy., L. C. Datz, 321-322 Hibernia Bldg.

MICHIGAN AUTOMOBILE ASSOCIATION
January 25–26, Detroit. Pres., E. A. Skae, Hammond Bldg.

MICHIGAN ENGINEERING SOCIETY
January 12–14, annual meeting, Lansing. Secy., Alba L. Holmes, 574 Wealthy Ave., Grand Rapids.

MONTANA SOCIETY OF ENGINEERS
January 6–8, annual meeting, Butte. Secy., Clinton H. Moore.

NATIONAL ASSOCIATION OF AUTOMOBILE MANUFACTURERS
January 12, annual meeting, Madison Square Garden, New York. Secy., Benjamin Briscoe, 7 E. 42d St.

NATIONAL ASSOCIATION OF CEMENT USERS
February 21–25, Chicago. Secy., R. L. Humphrey, Harrison Bldg., Philadelphia.

NATIONAL CIVIC FEDERATION CONFERENCE
January 5–7, Washington, D. C. Secy., D. L. Cease, 281 Fourth Ave., New York.

NEBRASKA CEMENT USERS ASSOCIATION
Februrary 1–4, Lincoln. Secy., Peter Palmer, Oakland.

NEW ENGLAND GAS ASSOCIATION
February 16, 17, annual meeting, Boston. Secy., N. W. Gifford, East Boston.

NEW ENGLAND WATER WORKS ASSOCIATION
January 12, annual meeting. Secy., Willard Kent, 715 Tremont Temple, Boston, Mass.

NORTHWESTERN ELECTRIC ASSOCIATION
January, Milwaukee, Wis. Secy., R. N. Kimball, Kenosha, Wis.

PACIFIC COAST ELECTRIC AUTOMOBILE ASSOCIATION
February, Oakland, Cal. Secy., J. T. Halloran, 604 Mission St., San Francisco.

RICHMOND RAILROAD CLUB
January 11. Lectures: Block Signals, Chas. Stephens; Terminal Freight Handling, G. H. Condict. Secy., F. O. Robinson.

SOUTH DAKOTA INDEPENDENT TELEPHONE ASSOCIATION
January 11–13, Huron. Secy., E. R. Buck, Hudson.

SOUTHERN GAS ASSOCIATION
February 16, Chattanooga, Tenn. Secy., James Ferrier, Rome, Ga.

STEVENS ENGINEERING SOCIETY
January 4, 11, 18, 4.10 p.m., Stevens Institute, Castle Point, Hoboken, N. J. Papers: Engineering Efficiency, H. G. Stott, Mem.Am.Soc.M.E.; Warfare of the Future, Hudson Maxim; Features of Electrical Development, T. C Martin. Secy., R. H. Upson.

WESTERN SOCIETY OF ENGINEERS
January 12, annual meeting, Chicago. Secy., J. H. Warder, 1735 Monadnock Blk.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
January			
1	Amer. Soc. Hungarian Engrs. and Archts.	Z. deNémeth	8.30
5	Wireless Institute	S. L. Williams	7.30
6	Blue Room Engineering Society	W. D. Sprague	8.00
11	The American Society Mech. Engrs	Calvin W. Rice	8.15
12	American Society Engrg. Contractors	D. J. Haner	7.30
13	Illuminating Engineering Society	P. S. Millar	8.00
14	American Institute Electrical Engineers	R. W. Pope	8.00
18–20	Heating and Ventilating Engineers	W. M. Mackay	All day
18	New York Telephone Society	T. H. Lawrence	8.00
21	New York Railroad Club	H. D. Vought	8.15
26	Municipal Engineers of New York	C. D. Pollock	8.15
February			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engrs. and Archts.	Z. de Németh	8.30
8	The American Society Mech. Engrs	Calvin W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15

OFFICERS AND COUNCIL

PRESIDENT

George Westinghouse ....Pittsburg, Pa.

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Geo. M. Bond ....Hartford, Conn.
R. C. Carperter ....Ithaca, N. Y.
F. M. Whyte ....New York

Terms expire at Annual Meeting of 1910

Charles Whiting Baker ....New York
W. F. M. Goss ....Urbana, Ill.
E. D. Meier ....New York

Terms expire at Annual Meeting of 1911

PAST PRESIDENTS

Members of the Council for 1910

John R. Freeman ....Providence, R. I.
Frederick W. Taylor ....Philadelphia, Pa.
F. R. Hutton ....New York
M. L. Holman ....St. Louis, Mo.
Jesse M. Smith ....New York

MANAGERS

Wm. L. Abbott ....Chicago, Ill.
Alex. C. Humphreys ....New York
Henry G. Stott ....New York

Terms expire at Annual Meeting of 1910

H. L. Gantt ....Pawtucket, R. I.
I. E. Moultrop ....Boston, Mass.
W. J. Sando ....Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

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James Hartness ....Springfield, Vt.
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STUDENT BRANCH	AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	....	C. F. Hirshfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	C. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	P. H. Van Etten	H. L. Hess
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. A. Kirk	J. R. Jackson
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	H. S. Coleman	John Garver
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	AndrewHamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	W. F. Colman	S. G. Wood
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	....	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Fredk. A. Dewey	A. P. Truette
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	H. B. Cook	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	....	....
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. E. Dudley	E. C. Phillips
Univ. of Nebraska, Lincoln, Neb.	December 7	....	....	....

THE ELECTRIFICATION OF TRUNK LINES

By L. R. Pomeroy, New York
Member of the Society

It is assumed from a physical and mechanical viewpoint, that electric traction can meet all the demands and requirements of railroad service. Therefore, whether electricity will replace steam traction or not is entirely a commercial problem.

THE COMMON DENOMINATOR—COMMERCIAL CONSIDERATIONS.[1]

2 It may be stated at the outset that whatever system of electrification is adopted, a very large outlay has to be faced and no case for electrification can be made out unless an increase in net receipts can be secured sufficient to more than pay interest on the extra capital involved. This increase may be brought about either by decreasing the working expenses for the same service, by so modifying the service as to bring in a greater revenue, or by a combination of these.

3 However, there is hardly a steam road in existence to-day which does not have divisions or sections, where distinctly local traffic can be handled more profitably by light, comparatively frequent electric service, than as now, with heavy steam trains. Both steam and electric service can be operated over the same tracks without detriment or embarrassment to either. In so doing each kind of

[1] Commenting on the problem of electrification of the Central Pacific over the Sierras, Mr. Kruttschnitt says: "Eastern critics may be inclined to the opinion that we are dallying with this matter. We have found that it pays well to make haste slowly with regard to innovations. Electrification for mountain traffic does not carry the same appeal that it did two years ago. Oil burning locomotives are solving the problem very satisfactorily. Each Mallet compound locomotive, having a horsepower in excess of 3,000, hauls as great a load as two of former types, burning 10 per cent less fuel and consuming 50 per cent less water."—*Wall Street Journal.*

The American Society of Mechanical Engineers.
All papers are subject to revision.

service would be appropriately handled in a manner best suited to the conditions of each.

4 The fundamental principle, based on the present state of the art, seems to be that if you cannot accomplish something by means of electricity that is now impossible by steam traction, there is nothing to justify the change; the mere substitution of one kind of power for another, merely to obtain the same result, is not commercially warranted.

5 There are certain inherent advantages in electrical operation that have shown up very well, because the increase in business has absorbed the increased interest account, but these cases hardly apply to trunk line conditions as the law of induced travel has no bearing on freight train operation, the principal business of trunk line roads.

6 In heavy work the limiting feature of the steam locomotive is the boiler, and the maximum adhesion can be utilized only at low speeds. For example, a 2-8-0 locomotive with 180,000 lb. on the drivers, has a tractive force, at 10 miles per hour, of about 40,000 lb. or 4.5 to 1. At 30 miles per hour the tractive force becomes 13,250 lb. or 30.2 to 1. As tractive force governs the tonnage hauled, the ability of the electric locomotive to utilize almost indefinitely power proportional to the maximum adhesion and produce a drawbar pull entirely independent of the critical speed of a steam locomotive, as limited by the boiler, is a marked feature.

7 In heavy grade work the ability to increase the speed shows up favorably to the electric locomotive as enlarging the capacity of a given section, but here also the business has to be sufficient to absorb the increase in fixed charges.

8 With steam locomotives a coal consumption, when running, of 4 to 5 lb. per i.h.p. hr. really means 6 or 7 lb. at the rail, when the losses due to firing up, laying by in yards and sidings, blowing off at the pops, and consumption of the air pumps, are taken into account. Whereas, under electric operation, with an efficiency of 65 to 70 per cent between the power house and the rail, a coal consumption of 4 lb. per kilowatt hour at the rail can be counted on.

9 The writer is informed that the Metropolitan Street Railway station (1903) with a 40 per cent load factor, produced power, at the switchboard, at the rate of 4.7 mills per kilowatt hour (or 3.5 mills per horsepower hour), and with a load factor of 55 per cent which prevails in the winter time, the cost is at the rate of 4.43 and 3.3 mills respectively. These costs cover all expenses and repairs except

fixed charges. The coal consumption is 2.9. lb. per kilowatt and 2.16 per horse-power hour.

10 L. B. Stillwell is authority for the statement that the Interborough is producing power at the rate of 2.6 lb. of coal per kilowatt hour or 3 lb. at the drawbar.

11 Another authority gives the following figures for the elevated roads for cost of power, \$0.005 per kilowatt hour at the switchboard, \$0.0066 at the third rail shoes, or \$0.0089 at the rims of the drivers. These figures are exceptional and hard to duplicate and as the fixed charges are not included, the writer would consider $1\frac{1}{4}$ cents per kilowatt hour at the rail a conservative figure, and will use this cost in the following computations.

RELATIVE COST OF COAL FOR STEAM AND ELECTRIC OPERATION

12 It may be fair to assume that where average coal is used, we can count on about \$2.25 per ton for locomotive coal on the tender, while a much cheaper grade can be used in the power house, costing, with modern coal handling facilities, about \$1.50 per ton. At this rate the relative difference in the cost of coal at the rail would be represented by the following figures:

Electric Power Station $\frac{2.5 \text{ lb}}{50\% \text{ off}} \times \$1.50$	\$7.50
Steam Locomotive 7 × \$2.25	\$15.75

or 50 per cent in favor of electricity. The following results of the Mersey Tunnel operation are pertinent: Under electric operation one ton of coal at \$2.10 yields 2.29 ton miles at $22\frac{1}{2}$ miles per hour, while with steam, one ton of coal, at \$3.84 yields 2.21 ton miles at $17\frac{3}{4}$ miles per hour. The difference amounting to 55 per cent is in favor of the electric operation, thus:

$$\left[1 - \frac{2.10}{3.84}\right] \times \frac{22.5}{2.29} \div \frac{17.75}{2.21} = \left[1 - \frac{2.10}{3.84}\right] \times \frac{22.5 \times 2.21}{2.29 \times 17.75} = 55\%$$

13 On mountain grades or in heavy freight service, where the boiler of the freight locomotive is forced to the limit, and the boilers are designed for this particular purpose, the showing is still more favorable to the electric side. Especially is this true when the steam locomotive is detained on side tracks for as long a period as it takes to make the run, which is very frequently the case, since under these conditions the cost for fuel becomes a larger proportion of the

total operating expense. A 2-8-0 locomotive with 50 sq. ft. of grate surface burns 300 lb. of coal per hour while lying on side tracks. Reports from Mallet locomotives indicate that from 600 to 800 lb. are burned per hour under the same conditions.

14 The cost of a unit of power with the steam locomotive becomes relatively higher under maximum than minimum boiler demands, while with electricity the cost per unit is at a uniform rate, whether working under extreme or light power demands.

For example:

15 *Case 1.* A consolidation (2-8-0) type locomotive with 180,000 lb. on 57 in. drivers, 50 sq. ft. of grate surface, working under maximum conditions on a $1\frac{1}{2}$ per cent grade, would burn 150 lb. of coal per sq. ft., of grate surface per hour and evaporate from 12 to 15 lb. of water per sq. ft. of heating surface per hour. Under these conditions the cost per 1,000 ton miles would figure out as follows:

$$\frac{F \times \text{price per ton} \times R \times 1000}{2000 \times \text{m.p.h.} \times E \times TF} = \text{Cost per 1,000 ton miles}$$

where F = coal per hour (150 lb. $\times$ 50 sq. ft. of grate surface).
R = resistance to be overcome [(grade per cent $\times$ 20) plus 6].
E = 80 per cent efficiency to cover losses such as cleaning fires, idle time while under steam, cylinder condensation, air pump consumption, etc.
TF = tractive force, in this case 180,000 lb. on drivers $\div$ 4.5 = 40,000 lb.

Substituting these values, the formula becomes

$$\frac{7{,}500 \text{ lb.} \times \$2.85 \times 36 \times 1{,}000}{2{,}000 \times 10 \times 80\% \times 40{,}000} = \$1.20$$

If the same service is handled by electric locomotives the cost on a similar basis becomes:

$$\frac{R \times \text{(watt hr. per ton mile)} \times 1{,}000 \text{ tons} \times \text{price per kw. at the rail}}{1{,}000 \text{ watts}}$$

$$= \frac{36 \times 2 \times 1{,}000 \times \$0.01\frac{1}{4}}{1{,}000} = \$0.90$$

17 If locomotive coal is taken at $1.70 per ton (the price in eastern Pennsylvania for low grade soft coal), the cost for coal for locomotives under the foregoing conditions would be:

$$(a) \quad \text{Steam, } \frac{\$1.20 \times 1.70}{2.85} = \$0.716$$

(*b*) Electric current reduced to 1c. per kw. hour at the rail:

$$\frac{0.90 \times 1\text{c.}}{1\tfrac{1}{4}\text{c}} = \$0.72$$

18 *Case 2.* An express passenger locomotive of the Atlantic (4-4-2) type, with the following data: Cylinders 21 by 26 in., boiler pressure 200 lb. per sq. in., weight on drivers 102,000 lb., heating surface 2,821 sq. ft., grate surface 50 sq. ft., rate of combustion 150 lb. per sq. ft. of grate surface per hour, speed 70 miles per hour. Figuring as in Case 1.

$$\frac{7{,}500 \times 2.85 \times 20 \times 1{,}000}{2{,}000 \times 70 \times 80\% \times 5{,}350} = \$0.71$$

Under electric conditions we have

$$\frac{20 \times 2 \times \$0.01\tfrac{1}{4} \times 1{,}000 \text{ tons}}{1{,}000 \text{ watts}} = \$0.50$$

or 28½ per cent less.

19 If coal is taken at $1.70 per ton, as in Case 1, the cost is reducd from $0.71 to $0.42, making the difference slightly in favor of steam.

20 These figures apply only to the conditions named, and average conditions on an undulating profile, when coasting is occasionally possible. With the benefits of momentum grades, also, the figures would be relatively less, but the electric locomotive would respond and benefit accordingly, so that the percentages would be approximately the same.

21 When steam locomotives are loaded to their capacity, as is generally the case where tonnage rating is practiced, the rate of combustion of 150 lb. of coal per square foot of grate surface per hour, will still hold good and remain constant, the tons hauled being the variable, responding or being modified by the speed or physical conditions of the road.

SAVINGS CLAIMED FOR ELECTRIFICATION

22 In view of the foregoing the following extract from an article by Mr. C. L. De Muralt will be of interest. The figures are from the annual report of 1903 of the roads named.

COST OF OPERATING TRUNK LINES

	P. R. R.	N. Y. C.
Fuel for locomotives	$6,000,135	$4,635,877
Water " "	335,286	295,583
Other supplies for locomotives	382,548	334,673
Wages: Engine men and roundhouse men	5,716,848	4,928,443
Other trainmen	4,442,127	2,991,335
Switchmen, flagmen and watchmen	3,900,427	2,511,552
Other expenses of conducting transportation	14,540,542	11,607,538
Repairs to locomotives	4,412,983	3,608,972
" other equipment	10,674,726	5,661,992
" roadbed	8,542,935	6,145,341
" structures	4,122,018	2,454,691
General expenses	1,858,319	1,786,494
	$64,928,894	$46,962,491

23 Mr. De Muralt then applies the figures found during the course of his investigation, which would lead to the following reductions if electricity was adopted as a motive power.

	P. R. R.	N. Y. C.
Fuel 10 per cent	$600,013	$463,388
Water saved entirely	335,286	295,583
Other supplies 50 per cent.	191,274	167,336
Wages, enginemen, etc., 25 per cent	1,429,212	1,207,361
Repairs to locomotives	2,206,492	1,804,486
Total amount saved	$4,762,277	$3,942,154

24 The saving in water alone capitalized at 5 per cent equals $6,750,000 for the former and nearly $6,000,000 for the latter road. As large as these alleged savings are, yet they would not amount to more than 2½ to 3 per cent on the necessary increase in capital to electrify the roads on which the foregoing savings apply.

25 While the first cost for power stations and electric equipment represents a large outlay, yet such items as the cost for repairs of locomotives and shops, expensive hostlering at terminals, coaling and water stations, and the incidental labor charge and repairs thereto will, in the aggregate, be materially reduced. The comparative saving in repairs will be indicated by the following figures:

Repairs	Steam	Electric
Boiler	20%	0%
Running gear	20%	20%
Machinery	30%	15%
Lagging and painting	12%	5%
Smoke box	5%	0
Tender	13%	0
	100%	40%

OTHER COMPARISONS BETWEEN STEAM AND ELECTRIC LOCOMOTIVES

26 It is further claimed that, with electric operation, greater mileage is possible with the electric locomotive and that fewer units are necessary to perform the same service. Great stress is laid on the fact that the ordinary freight locomotive makes only 3,000 miles per month, or 100 miles per day, against which is put forward

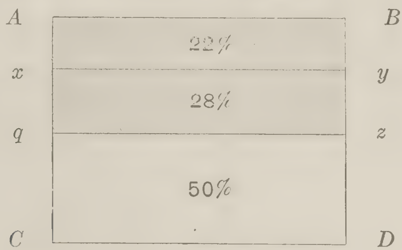


FIG. 1 DIAGRAM SHOWING DIVISIONS OF LOCOMOTIVE WORKING DAY

the ability of the electric locomotive to perform practically continuous service, suggesting the propriety of comparing electric and steam operation on the basis of ton miles per annum each is able to make and also the relative weight on driving wheels and not their total weight.

27 The operating efficiency of a steam locomotive in freight service is so low, averaging about 3,000 miles per month, that it is generally thought due to limitations, *per se*, in the locomotive, whereas it is mainly due to operating and traffic conditions, which limitations would apply with equal force to the electric locomotives, so that, barring some increase in speed, the electric locomotive can make no greater mileage than its steam competitor in equivalent service, consequently its splendid ability to perform almost continuous service cannot be realized in practice for reasons aforesaid.

28 Let the rectangle *A B C D* represent a day of 24 hours the shaded area *A B x y* that portion of the time for which the mechanical department is responsible = 22 per cent; the area *x y q z*, the average

time the locomotive is performing useful work = 28 per cent—*i. e.*, actually pulling trains, 3,000 miles per month, 100 miles per day; while the portion of the diagram bounded by $q\ z\ C\ D$, the period or balance of the time that the locomotive is under steam, with crew, and ready to go, and represents the time at terminal yards, side tracks and awaiting orders, etc. = 50 per cent.

29 It is just here that our electrical friends make the great mistake of claiming "greater capacity" for the electric locomotive over its steam equivalent. It is conceded that under electric conditions the area $A\ B\ x\ y$ may be reduced as much as one-half and perhaps, owing to greater speed, the area $x\ y\ q\ z$ may be increased, but the "lost motion" period due to traffic and operating causes will be relatively the same for both. The percentages are from an actual three months' test on a trunk line reported in1904 in the proceedings of the American Railway Master Mechanics Association by the committee on time service of locomotives.

30 The only cases where electric operation is commercially justified is in congested local passenger situations where the conditions closely approach those of a "moving sidewalk" and the records show that these cases have been profitable only when a large increase in business has been realized.

31 A modern Atlantic (4–4–2) type locomotive weighs, including tender, 321,620 lb. with a maximum tractive force of 23,500 lb. The ratio of total weight to tractive power is 133 to 1. The New York Central electric locomotive, with a total weight of 192,000 lb. and a tractive effort of 27,500 lb. has a ratio of 7 to 1. The comparison is still more favorable for electric freight locomotives where the entire weight is on the driving wheels.

POWER STATION CAPACITY

32 The impression is quite prevalent that if 100 steam locomotives are required to operate a certain division, if operated electrically, a power station capacity the equivalent of 100 locomotives would be necessary, whereas the generator capacity, barring the installation of spare units, would be of such size as to meet the average load. This average can be determined by laying down a train sheet, from which the load at any hour in the day can be seen and the peaks located.

33 For ordinary computations the number of trains to provide for is, approximately:

$$\frac{\text{The total train miles per hour}}{\text{Mean speed}}$$

This formula is the result of cancellation from the following:

(*a*) h. p. days ÷ Aggregate h. p.

That is:

$$(b)\quad \frac{5{,}280 \times (\text{Dis. miles}) \times (\text{No. trains}) \times (\text{Tons}) \times \text{R}}{47{,}520{,}000 \text{ ft. lbs. in 1 day}}$$

$$\div \frac{\text{Tons} \times \text{R} \times \text{m.p.h.}}{375}$$

R = resistance due to gravity, + resistance due to speed, + curve resistance. Transposing and cancelling:

$$(c)\quad \frac{\text{Dis. miles} \times \text{No. trains}}{24 \times \text{m.p.h.}}$$

For illustration take a typical case: Distance 183 miles.

LOAD		AVERAGE SPEED	
37 Freight Trains	at 15 m.p.h.	37 × 15 m.p.h. =	555
22 Expresses	at 50 m.p.h.	22 × 50 m.p.h. =	1,100
21 Locals	at 30 m.p.h.	21 × 30 m.p.h. =	630
80 Trains total.		80	2,285

2,285 ÷ 80 = 28 average m.p.h.

$$\frac{80 \text{ trains} \times 183 \text{ miles}}{24 \text{ hr.} \times 28 \text{ m.p.h.}} = 22 \text{ trains.}$$

34 For more accurate work a train sheet should be made either with miles as ordinates and time as abscissæ, or one with trains as ordinates on a time (abscissa) base.

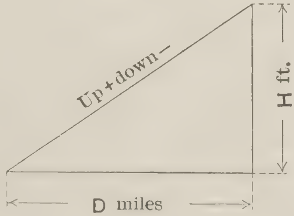
35 Relative to *R* (*i.e.*, resistance) for gravity; divide the profile into sections, one for each change in grade, plus or minus as thecase may be:

$$\frac{\text{H}}{\text{D} \times 52.8} = \text{Per cent grade.}$$

Each 1% grade = 20 lb. = R

R for curves 0.56 lbs. per degree.

$$\text{R for level sections} = 2 + \frac{\text{m.p.h.}}{4}$$



36 Consider the example of a road or division 100 miles long on which a given train requires 2,000 h.p. to keep it in motion. If 20

cars take a maximum of 100 h.p. each, the electrical conductors and distributing apparatus will never be required to deliver more than 100 h.p. at any one point. If on the other hand, the entire traffic of the line must be concentrated in a single train, the electrical conductors and distributing apparatus must deliver the full 2,000 h.p. at each and every point. In other words, with the concentrated load, the capacity of the distributing apparatus at each and every point must be 20 times as great as the capacity when 20 cars are used to give the same total load. Electric traction has proved its superiority for distributing loads, but concentrated loads are still handled almost exclusively by steam locomotives.

SOME ADVANTAGES OF ELECTRIC LOCOMOTIVES

37 In the annual report of the P. R. R. (1903) the president states "That the congested condition of your system has brought about a large increase in the ton mile cost, which for 1903 was 25 per cent greater than for 1899. In order to prevent the increase in ton mile cost, it is necessary to move freight trains faster in places where traffic is dense, and for such purposes the electric locomotive is most efficient."

38 With steam locomotives the most economical average speed, for freight service, is 12 to 15 miles per hour, where there is ample track space for the free movement of trains. With a dense traffic this free movement can only be obtained by a higher speed and if the large train tonnage be maintained, more horse-power is required of the engine and boiler. It is difficult to increase the size of steam freight locomotives without resorting to the Mallet compound articulated type, and here we have the equivalent of two locomotives in one machine.

39 With the electric locomotive it is possible to develop a much greater horse-power and a large percentage of overload at the time when needed and do it more economically than with steam. The New York Central electric locomotive has a maximum peak horse-power of 3,000, which is 25 per cent above normal. This maximum is about double the power which can be obtained from the New York Central standard Atlantic (4–4–2) type locomotive. Similar proportions can be obtained for electric freight locomotives and their size and power are not limited by boiler capacity. If the steam locomotive is capable of developing 30,000 T. F. at the drawbar at 12 m.p.h or

$$\frac{30{,}000 \times 12 \text{ m.p.h.}}{375} = 960 \text{ h.p.}$$

and it is required to increase the speed of the train to 20 m.p.h. and maintain the same tonnage, then 1,600 horse-power will be required, which means the employment of a much larger locomotive or double heading.

40 The advantage of the overload capacity on short mountain grades or for strategic peaks is one of the strong points in favor of the electric machine and would make electric operation applicable to special cases rather than a universal substitute, in the broad light of commercial considerations.

GENERAL CONCLUSIONS

41 Our conclusion, from this survey of the situation, is that the rapid development of suburban passenger traction by electricity will require large power houses at large cities and these can gradually be made sufficient for working the line on further stretches in each direction, handling congested terminals, or used where commercially practicable, until it may be desirable to electrify the entire division.

42 Electric operation as compared with steam shows to greatest advantage in urban and suburban passenger service. Here, if multiple unit trains are employed, so that a considerable fraction of the total weight is carried on the driving wheels, thus permitting a high rate of acceleration to be used, a schedule speed quite impracticable in steam operation can be maintained. Moreover, a more frequent service can be given without a proportional increase in expense, whilst in times of light traffic small trains can be run, the energy consumption per train in such service being almost in proportion to the number of coaches. The law of induced travel, however, applies to urban and suburban passenger service, but does not hold for trunk lines and especially freight service.

TO DETERMINE WHETHER IMPROVEMENTS ARE JUSTIFIABLE

43 Under trunk line conditions the only thing that interests railway managers is the traffic available at the present, relatively speaking; the future is too indefinite to be capitalized to any great degree in advance. It is more in the line of insurance companies to "capitalize expectations."

44 In grade revision the authorization for expenditure is based on the saving in train miles capitalized. The following is a concrete case from a Western road, or rather the summation of the engineers'

report as to just what the proposed rearrangement would amount to. The rate of 50 cents per train mile is to cover those items of cost directly affected by the change.

$$\left.\begin{array}{l}\text{No. of}\\ \text{trains per}\\ \text{day—7}\end{array}\right\} \times \left[1 - \frac{\text{1,350 tons present conditions}}{\text{1,600 tons proposed}}\right]$$

$$\times \left\{\begin{array}{c}\text{Div. of}\\ 225\\ \text{Miles}\end{array}\right\} \times 50\text{c.} \times \left\{\begin{array}{c}365\\ \text{days}\end{array}\right\} = \$45{,}990.$$

45 Under the circumstances it will be seen that the value of 1 per cent reduction in train mileage per mile per train, amounts to $1.95 per annum. The total amount capitalized at 5 per cent equals $919,800. In some such manner the steam railroad manager arranges the proposition of the electric scheme and decides accordingly.

SOME EXAMPLES

46 In a paper before the American Society of Civil Engineers by W. J. Wilgus, some interesting data concerning New York Central operation were given:

Cost of coal per 2,000 lbs. anthracite steam loco., terminal service.......	$4.46
" " bituminous coal, road service..................................	3.12
" " " " power station..........................	2.72
Water per 1,000 gallons:—	
Power station..	13.5 cts.
Road service..	5 "

47 The cost of current, when power station designed load is attained, is 2.6 cents per kilowatt hour delivered at contact shoes. This includes all operating and maintenance costs, interest on the electrical investment required to produce and deliver current, depreciation, taxes, insurance and transmission losses. The following table summarizes the data:

Items	Operating Costs	Fixed Charges	Total
Power Station	0.58 cts.	0.44 cts.	1.02 cts.
Transmission Losses	0.19 cts	0.15 cts.	0.34 cts.
Distribution Systems			
Substations	0.32 cts.	0.92 cts.	1.24 cts.
Totals............	1.09 cts.	1.51 cts.	2.60 cts.

48 In a discussion by G. R. Henderson (page 102, Vol. LXI, Trans. A. S. C. E.), are given road service costs per 1,000 car ton miles:

	Steam	Electric
Supplies	$2.03	$1.37
Wages	0.28	0.31
Interest, depreciation, and repairs to locomotive	0.46	0.34
	$2.77	$2.02

49 The item "Electric Supplies" is composed of operating expenses and fixed charges and may be analyzed thus:

53.3 kw. hour at $0.0109, $0.58 operation
52.3 kw. " " 0.0151, 0.79 fixed charges
52.3 kw. " " 0.026, 1.37

[Fixed charges = $\left(\frac{0.79}{1.37}\right)$ = 57 per cent of operating expenses]

The brackets are ours. The difference in cost between steam and electric traction in road service is $2.77 − 2.02 = $0.75 per 1,000 car ton miles.

50 The fixed charges on the power plant and the transmission system are $0.79 per 1,000 car ton miles, or about the same as the saving, so that if the train movement were but one-half the assumed amount (averaging 6,000 horse-power at the rails, or 6,000 kilowatts at the station) the cost for electric service would be slightly higher than for steam, or $2.81 as against $2.77 per 1,000 car ton miles.

51 The Manhattan Elevated, with about 38 miles of road, was electrified at an expense of $17,000,000. The operating ratio, under electric conditions, has been reduced from 61 to 46 per cent of gross receipts. The net result after taking care of the increased capital, etc., shows 15 per cent profit, but it is a significant fact that the increase in business was 46 per cent (carrying about 250,000,000 people per annum, 690,000 per day average, or 28,800 per hour).

52 There has just been reported the four years electric operating results of the Mersey tunnel road connecting Liverpool and Birkinhead. The net profit, allowing interest, etc., on the increased capital due to electrification, amounted to 15 per cent, but it took an increase in traffic of 55 per cent to make this operating result possible. Ton miles increased from 43 to 67 million, or 55 per cent. Total expenses, including interest on electric capital (but not depreciation) equal $0.586 per ton mile. Interest equals $0.106 per ton mile, or 22 per cent of operating expenses.

53 President Harahan of the Illinois Central reports the results

of the investigation that has been made relative to the proposed electrification in the following words:

54 "Our suburban traffic is the only service which would in any degree be adapted to electric operation, but even in this particular service it can be readily shown to be unjustifiable at the present time. I submit below a statement of the results which are estimated to accrue if the entire suburban service were electrified, compared with the present steam operation:

"Results of Operation of Suburban Business at Chicago for Fiscal Year ending June 30, 1909:

Gross earnings		$1,056,446
Operating expenses (82.9%) plus taxes		946,734
Net revenue		$109,712
"Estimated Results Under Electrification:		
Gross earnings		$1,056,446
Operating expenses (66%)	$697,254	
Taxes	74,427	
		$771,681
Net revenue (electric operation)		$284,765
Net revenue (steam operation)		109,712
Increase		$175,053
Estimated cost of electrification		$8,000,000
Interest and depreciation 10%		$800,000
Saving in operation under electrification		175,003
Deficit		$624,947

55 "Our suburban traffic is not sufficiently dense to warrant the expense necessary to electrify these lines, and it is evident from the foregoing figures that even under electrification there would not be an increase in traffic sufficiently large to offset the annual loss from operation. It simply proves that under present conditions of cost of electrification of steam railways, where it means a replacement of a plant already installed, and serving the purpose, it is not justifiable to electrify either in whole or in part your Chicago terminals at this time."

56 The surburban district of the Illinois Central covers about 50 miles of road and carries in round numbers 15,000,000 surburban passengers per annum, or an average of 41,150 per day, or 1,700 per hour. An increase of 100 per cent in earnings would not enable the road to break even.

57 The Railway Age Gazette, in commenting editorially on Mr. Harahan's statement, says:

58 "It may be accepted as conclusively demonstrated that the New York Central and the New Haven roads are moving trains by electricity more economically than they moved them by steam in their suburban district. To enable this to be brought about, however, extremely heavy capital costs had to be assumed and the charges on these capital costs make the entire operating cost, including overhead charge, far higher than it used to be in the days of steam operation.

59 "For example, a standard express train of eight cars on the New Haven road pulls out of Grand Central station headed by two half-unit electric locomotives, each of which cost very nearly $40,000. The capital cost of the motive power of this train is in excess of $75,000 [the interest and depreciation amounting to $20 per day]—the brackets are ours. The cost of motive power at the head of a similar New York Central passenger train operated by electricity is about one-half this sum. Moreover, it will be recalled that Mr. Wilgus estimated that the direct costs of electrical equipment represented only one-fourth of the total charges attendant upon electricity. The cost of making everything ready and safe for this kind of operation is far greater than the highest estimates are apt to contemplate."

60 From a report of the Electrical Commission of the State of Massachusetts the following extracts are taken (letter of C. S. Mellen, president of the New Haven road):

61 "We believe we are warranted in saying that our electric installation is a success from the standpoint of handling the business in question efficiently and with reasonable satisfaction, and we believe we have arrived at the point where we can truthfully say that the interruptions to our service are no greater, nor more frequent, than was the case when steam was in use. But we are not prepared to state that there is any economy in the substitution of electrical traction for steam; on the contrary, we believe the expense is very much greater."

62 The Boston & Albany Railroad Company reports the result of their study and estimates the requirements as follows: A power station of 6000 kilowatts will be necessary, with storage batteries to handle the peak load. The total cost of the installation is estimated at $4,000,000, and the interest, taxes, and depreciation at 9 per cent, or about $400,000 per annum. A stock argument for electric operation is the saving to be made in operating expenses, but concerning this the following statement is made:

63 "Some slight economies would accrue in the transportation

expenses under this operation, which would be substantially absorbed by the additional expenses to be incurred for the maintenence of the additional apparatus installed, and the net economies would be so small as to be inappreciable in the consideration."

64 Another stock argument of the advocates of electric locomotives is the growth of traffic which is supposed to result from electric operation. This argument is met as follows in the report:

65 "Considering now the possibilities of increasing the traffic, the statistics of the B. & A. R. R. show substantially the following number of passengers handled in the above territory per annum:

1891	4,552,918	1899	3,897,364
1894	4,799,578	1907	4,435,841

66 "The absence of any material increase in traffic is probably due to the fact that the circuit is occupied as a high class residential district not susceptible of rapid subdivision of property, and more particularly to the fact that suburban lines are being rapidly extended into all such outlying districts and afford a more advantageous means of collecting and distributing local travel through the commercial and residential districts than could possibly be afforded by a railroad constructed and operated upon private right of way and devoted largely to long haul operations."

EXAMPLE TO ILLUSTRATE A CONCRETE CASE

67 The following illustration representing a concrete case is selected because of its elementary character, more especially as the case is so simple that all the variables affecting the comparison are eliminated and the amount of coal to perform the operation is directly known:

Conditions: trailing load 1,600 tons; average grade, 1.3 per cent.; distance, 8 miles; speed, 15 miles per hr. for electric and 14 miles per hr. for steam locomotive.

(*a*) Electric

1,600 net tons
190 Loco. (2) tons
——
1,790 gross tons

$$R = \begin{cases} 1.3\% \text{ grade} \times 20 = 26 \text{ lb.} \\ 5° \text{ curves} \quad 3 \text{ lb.} \\ \text{Level} \quad 6 \text{ lb.} \end{cases}$$

$$\frac{\text{Gross tons} \times R \times \text{Distance}}{500} = \text{kw-hr. at the rail}$$

Substituting values:

$$\frac{1{,}790 \times 35 \times 8}{500} = 1{,}000 \text{ kw-hr. (at rail)}$$

Equivalent kilowatt load at power house =

$$\frac{\text{Tons} \times R \times \text{ m.p.h.}}{500 \times \text{Efficiency } \%}$$

Where the efficiency between the rail and generators equals 65 %, substituting as before:

$$\frac{1{,}790 \times 35 \times 15}{500 \times 65\%} = 2{,}900 \text{ kw.}$$

For this particular case current can be purchased from an adjacent power house at the very low rate of one cent per kw-hr. at the rail.

At this rate the power cost per trip will be 1,000 kw. at one cent =\$10.00.

(*b*) Under steam conditions we have the same as before, 1,600 net tons + weight of two locomotives, 300, or 1,900 gross tons.

The coal consumption for this particular run is 6,000 lb.

The price per ton to equal the electric cost for power, is:

$$\frac{6{,}000 \text{ lb.} \times \text{price per ton}}{2{,}000} = \$10.00$$

Transposing:

$$\frac{2{,}000 \times 10}{6{,}000} \times \$3.33$$

But as coal for this particular case costs the road \$1.70 per ton, the relative cost, coal against power, is

$$\frac{6{,}000 \times \$1.70}{2{,}000} = \$5.10$$

There is a difference in ton mile hours, in favor of the electric locomotive, due to speed and reduced gross tonnage, as follows:

$$\text{1st Electric } \frac{1{,}790 \times 8 \times 8}{15} = 7{,}640 \text{ Gross ton mile hours}$$

$$\text{2d Steam } \frac{1{,}900 \times 8 \times 8}{14} = 8{,}690 \text{ Gross ton mile hours}$$

To make the comparison correct the coal consumption of the steam locomotive should be proportioned on the ton mile hours, produced, and the cost of coal then becomes:

$$\frac{\$5.10 \times 8{,}690}{7{,}640} = \$5.80$$

Adding to the foregoing the other operating costs the relative expense becomes

(*a*) *Electric.* Power.. \$10.00

Lubrication, supplies, repairs, crew at \$0.1158 per 1,000 ton miles, or

$$\frac{0.1158 \times 1{,}790 \times 8}{1{,}000} = \text{..................} \quad 1.66$$

Interest and depreciation, taxes, insurance, etc., at 10%.. 1.46

\$13.12

(*b*) *Steam.* Coal as above	$5.80
Lubrication, supplies, water, repairs, enginemen at $0.25 per 1,000 ton miles,	
$\frac{\$0.25 \times 1{,}900 \times 8 \text{ miles}}{1{,}000} =$	3.80
Interest and depreciation at 10 % (2 locomotives)	
$\frac{\$34{,}000 \times 10\% \times 8}{365 \times 24 \times 14} =$	0.22
	$9.82

Cost per trip in favor of Steam, $3.30, or 25 % less

EXAMPLES TO ILLUSTRATE A CONCRETE CASE

68 The idea is all too prevalent with the public, and even with some of the bodies that have been given legal power of supervision over railway companies, that any expenditure which can be forced upon the railway companies is just so much gain for the public. Never was there a more absolute fallacy. In the long run, the cost of every bit of railway improvement must be paid for by those who buy tickets and ship freight. Economy in the administration of our railways is just as important in the interest of the general public as if the railways were actually under government ownership.

Recently *The Engineer* (London) editorially made a plea for a "common denominator" for comparison of engineering achievements, using the following illustrations:

"Thus for example, if we take Mr. Humphrey's reply to Mr. Davey's criticisms, we see that he gained a mere dialectical advantage by showing on the screen a great differential pump, and beside it an internal combustion pump, so small by comparison that he had to explain that it was not a "hooter." Both engines could deal with the same quantity of water; but the Davey engine was lifting it 1500 ft. from a mine, while the gas pump could not lift it more than about 15 ft. Indeed, it could not do the work of the Davey engine at all."

Also a comparison was drawn between the cost of working with producer gas engines and steam engines. The argument was all in favor of the gas engine, expressed in weight of fuel required per hour to develop a horse-power. But the aspect of the matter changed when it was pointed out that the coal used by the steam engine was slack, costing $1.75 per ton, while the gas producer worked with anthracite, costing over $6.25 a ton. Here the cost of fuel was the common denominator, not the weight of the fuel.

The plea concluded by saying that the common denominator should be the commercial cost. E. H. McHenry expressed the same idea when he said that "Engineering is making a dollar earn the most interest."

LUBRICATION AND LUBRICANTS

By Charles F. Mabery[1]
Non-Member

Next to the conservation of the world's fuel supply there is probably no subject of greater importance in the manufacturing world than the control of waste power caused by imperfect lubrication and needless friction. Notwithstanding the increasing interest in more economical methods, the immense losses from this source are scarcely appreciated. In his recent work on lubrication and lubricants, Archbutt stated that with considerably more than half the 10,000,000 h.p. in use in the United Kingdom of Great Britain, 40 to 80 per cent of the fuel is spent in overcoming friction, and that a considerable proportion of this power is wasted by imperfect or faulty lubrication. On account of the great abundance of cheap fuel in the United States doubtless the conditions here are even less desirable. It is safe to state that losses from this source in this country are from 10 to 50 per cent of the power employed. Not infrequently in factories where the annual expense for lubrication amounts to thousands of dollars, lubrication experts find a loss of 50 per cent, or greater.

2 The manufacturer often knows very little concerning the economic qualities of the lubricants he receives; in using them, too much is left to "rule-of-thumb" methods with little knowledge of the actual conditions of friction, the action of metallic surfaces under the dynamic stress of the transference of power, or such modified action as is produced by the intervention of a lubricating film. For example, the different effects on a journal of a soft and hard bearing may be sufficient to cause a considerable loss of power if improperly selected, and yet may escape attention. In the earlier tentative study of the conditions depended on for the results described in this paper, under such loads as 100, or 150 lb. per sq. in. of bearing surface, the grades of babbitt in ordinary use were found much too soft and yielding to sus-

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tain such work under the necessary conditions of speed and oil feed; only a very hard alloy of exceptional composition could be used. The one selected of approximately the composition, Tin, 90; Copper, 2; Antimony, 8, gave results entirely satisfactory. Then since it was desired to maintain such conditions of load and speed that any oil could be broken down at any moment, it was found necessary, not only that the journal and bearing be milled to mechanically true surfaces, but that by continued operation and repeated careful milling, even a higher degree of permanent evenness be maintained. If such be the essential conditions in precise quantitative observations, similar precautions are evidently necessary in factory operations.

3 In the earlier days of machinery lubrication before the introduction into the trade of products from petroleum, the manufacturer had little concern about viscosity and other physical constants of lubricants, for, dealing with simple oils or greases of definite composition, he could be sure of obtaining what he desired within the capacity of the materials at his disposal. Then, in the days of higher prices of manufactured products and less severe competition, imperfect lubrication was of less consequence than in more recent times when every detail of cost and loss should properly receive careful attention; and, furthermore, the principles of friction and the importance of its control were only imperfectly understood in the earlier days of lubrication. Modern high speeds and excessively heavy loads had not then to be provided for in the applications of power in manufacturing operations, or in transmission or transportation.

4 The discovery that the heavy hydrocarbons in petroleum possessed the qualities requisite for lubrication—viscosity, durability and stability under varying conditions of speed and load—was the beginning of a new era in lubrication. Methods of treatment and refining, with little or no knowledge of the hydrocarbons of which the lubricating oils were composed, and developed entirely along empirical lines, were slow in producing suitable products. The earlier methods have undergone no fundamental changes even to the present time, except in the introduction of heavier hydrocarbons from crude oil territory more recently developed. Crude oils of the Pennsylvania type containing a considerable proportion of the hydrocarbons $C_n H_{2n+2}$ have always yielded excellent light spindle oils composed for the most part of the hydrocarbons, $C_n H_{2n}$, and $C_n H_{2n-2}$. But, as we now know, this type of oils include too small a proportion of the heavier hydrocarbons for the body necessary in lubricants subjected to the great stress of heavy loads and cylinder friction. This need in heavy

lubrication led to the practice of compounding oils, or mixing with the petroleum products various proportions of the vegetable oils, such as castor or rape, and the various animal oils or greases, which so fully monopolized this field, that manufacturers were often led to believe that no other products could serve an equivalent purpose. Even since the more recent introduction of heavy lubricants from Texas and California petroleum the belief still prevails that only compounded oils can be relied on for heavy work. But with care in distillation and treatment, it is certain that heavy lubricants, well adapted for bearings and cylinders, may be prepared from those crude oils, and large quantities of such lubricants are now widely in use.

5 All experimenters with lubricating oils who have given thoughtful attention to the essential needs of lubrication have been impressed by the superiority of an ideal solid lubricant, i. e., one that should embody an equivalent of the desirable qualities of the liquid products with a greatly superior wearing quality, a low coefficient of friction, and readily convertible into a form that can conveniently be applied to the various forms of journals and bearings. Soapstone, asbestos, natural graphite, etc., do not, altogether, possess these fundamental qualities of the liquid products. Greases compounded with graphite are useful on low-speed bearings and under heavy work. Natural graphite serves an excellent purpose on cast-iron bearings, acting as a surface evener of the porous metal. On finer surfaces care is necessary that it does not collect in such quantities as seriously to scratch or abraid the journal and bearing.

6 Of all the solid bodies available for lubrication, graphite possesses the desirable unctuous quality and great durability. For general use in lubrication, graphite must be in its purest condition and in a state of extreme subdivision. Whether, in such a condition as the deflocculated form, the ultimate molecules or atoms have a certain freedom of movement, analogous to that of liquid molecules under stress of friction, or whatever explanation may be suggested of its unctuous quality, the fact remains that it possesses this quality in very high degree. Such graphite is now produced by processes discovered, perfected, and placed on a manufacturing basis by Dr. Edward G. Acheson of Niagara Falls as a part of his great work in the development of electrochemical processes. Besides his immense output of pure graphite for general commercial use, Dr. Acheson has succeeded in converting it into a new form, a deflocculated condition, that meets the requirements of an ideal solid lubricant. This deflocculated form greatly surpasses ordinary graphite in unctuous quality,

and its adaptability for prolonged suspension in water and oils renders it especially applicable to frictional conditions. Furthermore, the readiness with which it forms coherent films on journals, its great wearing qualities and the ease of the application, constitute a lubricant of extremely high efficiency.

7 Acheson graphite can be produced from any substance that contains carbon in a non-volatile form. Under the extreme temperatures of the electric furnace any and all other elements are readily volatilized. Even carbon itself is freely vaporized and its peculiar appearance in the burning carbon-monoxid is depended on as an indicator of suitable conditions in furnace operation, much as the drop in the manganese flame which shows the disappearance of carbon in the Bessemer converter.

8 As commercial products two forms of graphite are produced, the unctuous and the deflocculated modifications, the first form accompanying the production of carborundum in furnaces charged with carbon and sand, the second obtained from a charge of coal or coke alone. The first form is leafy in structure, coherent, and extremely unctuous or greasy to the touch; it is segregated and not readily disintegrated. The second form is also unctuous in a high degree, but very pulverulent and capable of extreme subdivision; it is readily converted into a deflocculated condition. This form in water forms the commercial "Aquedag," or aqueous Acheson deflocculated graphite. In combination with oils it is known as "Oildag."

9 This deflocculated graphite has peculiar properties; it remains suspended indefinitely in water, but is quickly precipitated by impurities. On account of its extreme subdivision a very small amount suspended in water serves for efficient lubrication. From numerous and long-continued trials it appears that 0.35 per cent serves an adequate purpose and that a larger proportion is superfluous. It is certainly remarkable that such a small quantity of graphite is readily distributed by water between a journal and bearing while sustaining a load of 70 lb. per sq. in. of bearing surface, and that under high-speed conditions it maintains an extremely low coefficient of friction.

10 Proper lubrication of bearing surfaces involves careful consideration of the metals composing the journal and bearing, since the influence of the metals employed has an effect even in the intervention of the best lubricating film. The materials in common use for the construction of bearings include cast iron, steel, and alloys of variable composition included under the general terms, bronze and babbitt. In high-speed work cast-iron bearings must be used with extreme care.

In the accurate adjustment necessary in machine testing of lubricants, we have found it impossible to prevent injury to the journal when using a cast-iron bearing. Results obtained by the use of bronze have not been altogether satisfactory. However, properly selected babbitt for a steel journal seems to fulfill the desired conditions most satisfactorily and it possesses a wide range of applicability. As mentioned above, satisfactory lubrication is possible only when the journal and bearing are properly machined to true surfaces, kept smooth, accidental scratches worked out, and bare spots avoided. Successful lubrication demands constant skilled attention to the condition of journals and bearings, and no factory supervision affords more desirable returns. Lubrication consists in reducing friction to the lowest increment of the power in use. A lubricant is an unctuous body that readily forms a continuous, coherent, durable film capable of holding apart rolling or sliding surfaces, and itself interposing the least possible resistance. The economic problem in lubrication depends on the use of such a lubricant under suitable conditions.

11 The lubricants in commercial use include water, oils, greases and solids. Under oils are classified the great variety of light spindle, heavy engine and cylinder products, either unmixed hydrocarbons from petroleum or compounded oils, tallow, wool grease, etc. The greases may be generally classified under a few heads depending on their consistency, which is derived from the proportion of lime or soda soaps or oleates mixed with the hydrocarbon oil as a carrier. The solid greases have already been referred to.

12 Water in itself possesses no oiliness whatever but under certain conditions in cylinders it is found to assist in imparting to the metallic surfaces an extremely smooth condition which serves materially to reduce the friction. A practical knowledge of hydrocarbon lubricants should include a knowledge of the source, that is, the crude oil from which the lubricant is prepared, since there is a wide difference in composition and properties of the oils from different oil fields. Methods of refining petroleum oils have very much to do with the quality of the products. In general terms, inferior products are obtained when the process of distillation is conducted in such a way as to produce decomposition; the best products are obtained only by careful distillation and careful treatment in refining, whereby the hydrocarbons in the refined products obtained have essentially the same composition as in the original crude oil.

13 An examination of various lubricants in the trade frequently reveals a condition of the oils indicating improper refining. For

example, it does not need the application of extremely delicate tests to show the presence of free alkali, of sodium sulphate, or of sodium salts of organic acids, any one or all of which may be injurious to metallic surfaces. One of the most exacting duties of the refiner is the treatment with caustic soda in such a manner as to remove all acid products and at the same time to avoid such an excess of caustic as will form an emulsion, which is one of the "terrors" in a refinery. An examination of a great variety of oils in the trade, such for instance as the spindle oils in use in automobile service, indicates that the best refined oils are those that contain a minute trace of alkali.

14 The ordinary methods of testing lubricating oils include determinations of the viscosity, the specific gravity, the flash and the fire temperatures. Another important property of these oils, termed oiliness or greasiness, is not so readily determined by analysis; in fact, there seems to be no accurate method for its determination; yet it is readily distinguishable and has much to do with the efficiency of all lubricating oils. Concerning the most efficient methods of testing lubricating oils various opinions are expressed by different authors. Redwood, in his work on petroleum and its products, asserted that the viscosity of an oil is the best guide to its lubricating value since it enables the consumer to select oils similar to those that have afforded him the best practical results. He alludes to the close relationship between viscosity and the laws of friction of liquids. In comparing the use of viscosity with observations on the behavior of lubricants on a frictional testing machine, he states that he was unable to obtain satisfactory results with any machine at his disposal. His conclusions in general were that in the present state of our knowledge the indications afforded by testing machines are wholly misleading, and this led him to attach especial importance to a good system of testing viscosity. He refers to the opinion of Thurston that any oil should be tested on a machine under the conditions of load and speed similar to those of the use for which the oil is intended.

15 Referring to the work of Ordway and Woodbury in 1884 with an apparatus constructed to apply pressures of 40 lb. per sq. in.; to those of Tower carried on under what he terms great pressures—100 to 600 lb. per sq. in.— in an oil-bath system of lubrication; and also referring to the opinions of others on these results, Redwood presents the view that the agreement between machines and actual practice is extremely slight, his final conclusion being that viscosity affords the most valuable tests of lubricating qualities at our disposal. Inasmuch as Redwood's opinion on machine testing is a result of his observations

during several months on the Ingram and Stafer machines, in which the speed is 1500 r.p.m., and that the friction is gaged by the number of revolutions necessary to carry the temperature to 300 deg. fahr.: it is not difficult to understand his conviction that in his experience testing machines do not afford results comparable with those of actual practice.

16 The value of viscosity as a distinguishing property of lubricating oil is recognized by all who have given attention to the subject, but all are not agreed as to the extent of its practical reliability. Archbutt suggests that the quality of oiliness or greasiness is nearly of as much importance as viscosity. Although, as mentioned above, there is no precise method whereby oiliness can be determined, it is not difficult to recognize it nor to distinguish the marked differences in this respect shown by different oils and greases. Archbutt calls attention to the fact that at very low speeds the friction of a cylindrical journal should be proportional to the viscosity of the oil; but at higher speeds, and consequently increased temperatures, the relation of friction to speed ceases; the viscosity is diminished with a corresponding change in the carrying power of the journal. While fully appreciating the value of the information to be obtained by chemical analysis, Archbutt insists that the oiliness of a lubricant is of especial importance under heavy loads and high speeds. He suggests that it is advantageous for an engineer to test oils for himself on a machine without depending altogether on analytical data of physical tests obtained from the expert. Hurst also mentions that a broader knowledge of the practical working of oils is necessary than can be obtained from chemical or physical tests alone. He maintains that the test of an oil from a journal under the practical conditions of its use show conclusively its adaptability to such use.

17 The principal points to be observed in mechanical tests are the effects of speed, load, temperature, and the frictional effects due to viscosity and oiliness. The measurements on which depend the quality of the oil include the frictional resistance, the temperatures, and the endurance of the oil film. Doubtless the numerous machines that have been constructed for testing oils have certain merits and advantages. In the wide range of work carried on in this field during the past year, a part of the results of which are presented in this paper, the machine devised by Professor Carpenter was used. In its sensitive adjustment, durable efficiency, and the wide range of possible tests, this machine in continuous use during this period on light and heavy oils, greases and graphite, has fulfilled all requirements. Since

the results to be presented are closely dependent upon the method employed a view of this machine is here introduced.

18 This machine has an accurate adjustment for recording the speed, and a long lever arm with a vernier attachment graduated to tenths of a pound for recording the friction. The load is applied by a powerful spring worked by a cam and lever, the limit of the machine being 6000 lb., total load. Careful calibration of the spring showed it to be properly adjusted.

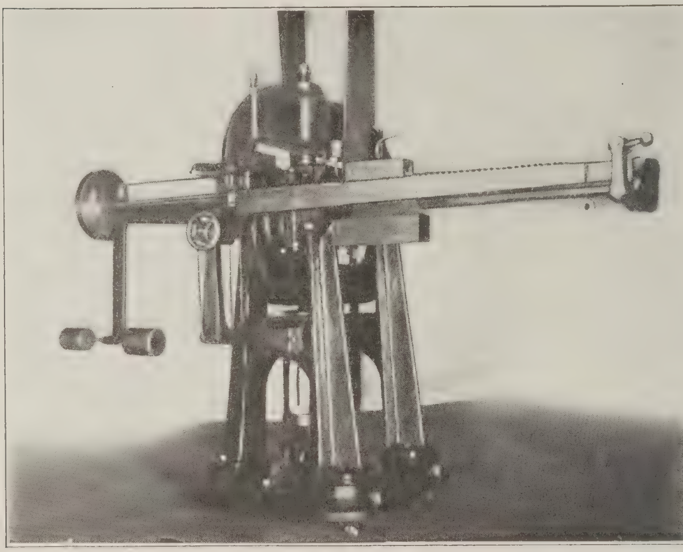


Fig 1 The Carpenter Machine for Testing Lubricants

19 In projected area the bearing in use is approximately 8 sq. in.; the journal is about 3 in. in circumference, nearly equal to 1 ft. in linear extension. A cast-iron frame babbitted and machined down to a true surface was used for the most part in this work. Even after careful machining some continuous frictional work was necessary on the babbitt surface to bring it to the proper conditions of constant results. The hard form of babbitt mentioned above gave satisfactory results, and there was little difficulty in keeping the surfaces in suitable condition after they were once obtained. For measuring temperatures a thermometer was inserted in a hole in the bearing, extending close to the journal.

20 Tests made at steam temperature—210 deg. fahr.—were carried on in a hollow cast-iron babbitted bearing, with steam attachments by which it was found that the desired temperature could readily be maintained. The lubricant is run in from a sight-feed cup through a small hole close to one side of the bearing with careful regulation of the flow for proper adjustment of the oil feed.

21 For delivery of the lubricant over the entire face of the bearing two channels or grooves are run diagonally across the babbitt face from the inlet hole, giving equal and even distribution; these channels must be carefully gaged for an even flow, otherwise dry spots or streaks appear on the journal accompanied by a sudden greatly increased friction indicated on the friction bar. This detail of operation requires careful and constant attention, for on it depends the continuous regularity of the friction curve. In this respect this method of observation is extremely sensitive, and is one of the important elements in frictional tests. Partial exposure of the journal enables the operator to observe the formation of the film, its comparative thickness and any irregularity due to an imperfect condition of the journal or bearing, or improper lubrication.

22 Accurate testing of the mechanical efficiency of oils with the precise quantitative observations possible on the Carpenter machine, including the various classes of lubricants under consideration in this paper, presented an extensive field of labor, especially since there are no general standards of comparison under any conditions of operation. Such constants must of necessity be based on arbitrary data, nevertheless if they are accurately determined on a standard machine, with the conditions of the journal and bearing selected,—the load and speed,—the constants on this machine may be readily ascertained on any other equally efficient machine. In duplicate tests made with the same bearing and under the same conditions the results were closely concordant. At the outset it should be clearly understood that these tests must be performed with a scientific accuracy of exact quantitative observations with close supervision of all details. The work then becomes the regular routine of any scientific investigation which involves long series of observations, after it is ascertained by preliminary trial what conditions are necessary in testing any given oil. Of course for commercial benefit these conditions should be as close as is practicable to the factory conditions of use.

23 The results to be described of the use of water, kerosene, and fuel oil as vehicles of graphite, present novel and interesting features. Under certain conditions, as mentioned above in steam cylinders, it is

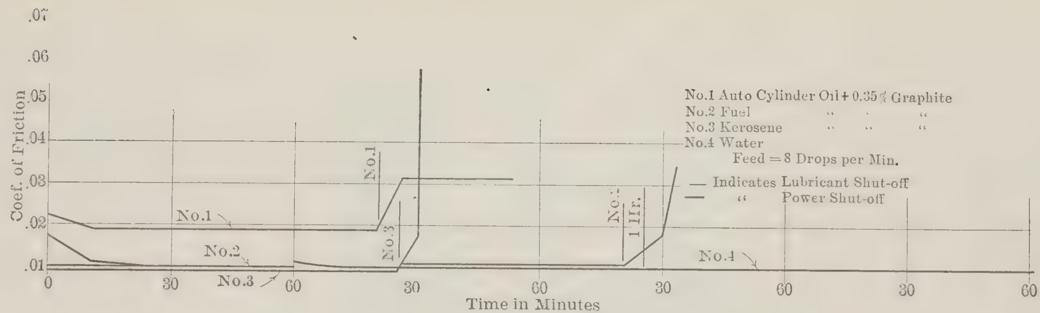


FIG. 2 CURVES OF FRICTION WITH VARYING VISCOSITY OF LUBRICANT. PRESSURE 70 LB. PER SQ. IN.; R. P. M. 446. NO. 1 AUTOMOBILE CYLINDER OIL + 0.35 PER CENT GRAPHITE; NO. 2 FUEL; NO. 3 KEROSENE; NO. 4 WATER. FEED = 8 DROPS PER MINUTE.

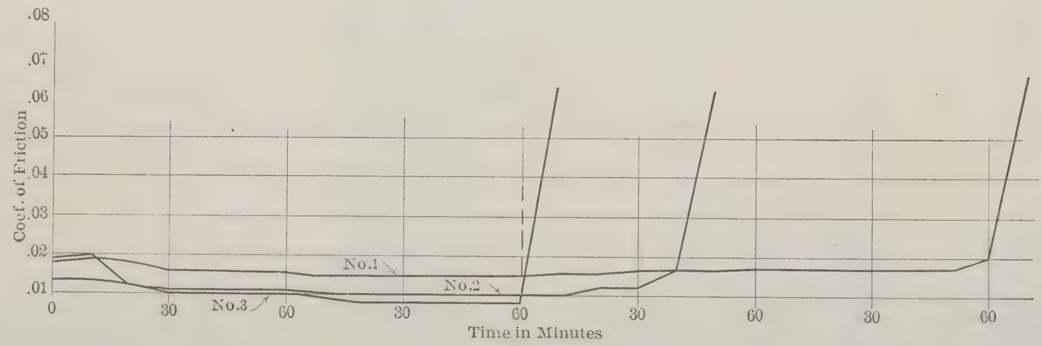


FIG. 3 CURVES OF FRICTION WITH VARYING VISCOSITY OF LUBRICANT. PRESSURE 150 LB. PER SQ. IN.; R. P. M. 445. NO. 1 FAILS SPINDLE OIL + 0.35 PER CENT GRAPHITE; NO. 2 FUEL; NO. 3 KEROSENE. FEED = 8 DROPS PER MINUTE. DOTTED LINE INDICATES LUBRICANT SHUT OFF.

well known to engineers that water alone serves as a lubricating film. But since on journals it serves no purpose whatever, the lubricating qualities of aqueous suspended graphite must be due wholly to the graphite. The same is true of kerosene, which alone is practically devoid of lubricating quality, and likewise of fuel oils.

24 For the purpose of testing the effect of varying viscosity in lubricants, and at the same time the lubricating quality of deflocculated graphite, tests were made with water, kerosene oil, a fuel oil, and an automobile cylinder oil, each carrying 0.35 per cent graphite. The results obtained in these tests are shown by the curves in Fig. 2, in which the speed is maintained at 446, and the load at 70 lb. per sq. in. The observations of frictional load and temperature were made at intervals of ten minutes and on that basis a curve is drawn for each of the lubricants tested; on the chart the time is given in half-hour limits and the coefficient of friction in hundredths of a unit. It will be observed that the curve for water and graphite is practically a straight line, with scarcely any variation for the four hours shown on the curve; this test continued for 15 hours altogether with a precisely similar result. There were several stops which are indicated by a dotted line on the chart, and it appears that there was no change whatever in the direction of the curve by stopping and starting. Curve 3, representing the observations on the coefficient for kerosene oil with graphite, is also a straight line showing a coefficient very slightly lower than water. The coefficient curve for the fuel oil and graphite is also practically a straight line, and with an endurance test extending 1½ hours after the oil supply was shut off; here the frictional coefficient is slightly higher than that of either water or kerosene. A similar regularity appears in the curve of the automobile cylinder oil with graphite; but it is to be noted that the frictional coefficient is very materially higher than those of the other lubricant shown on the chart, which may be considered as a measure of the comparatively greater internal viscosity of the automobile oil; this oil shows a much longer endurance test than appears on this chart.

25 The effect of varying viscosity in lubricants, and the lubricating quality of the graphite under practically the same speed, 445 r.p.m., but with a load of 150 lb. per sq. in.; using kerosene, a fuel oil and a spindle oil, with the same proportion of graphite, and the same oil supply, are shown on Fig. 3. Kerosene here shows a very slight irregularity in its coefficient, which differs only slightly from that on the preceding chart. Here again the greater internal viscosity of fuel oil is shown by the increased friction which appears in this curve. No

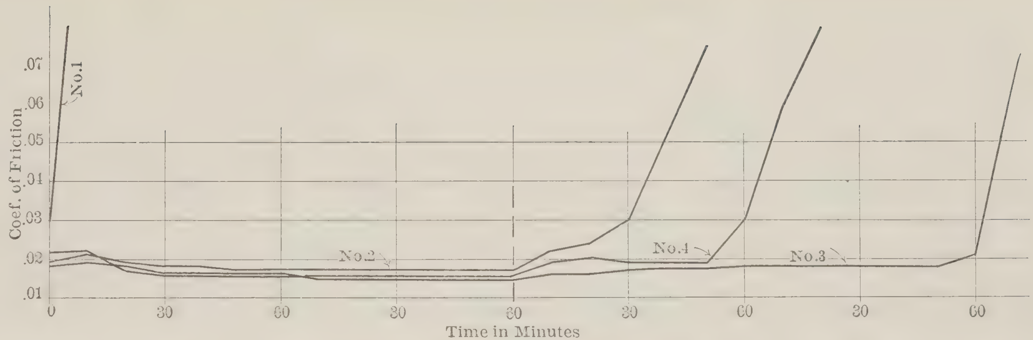


FIG. 4 CURVES OF FRICTION—OIL AND OILDAG—VARYING FEEDS. PRESSURE 150 LB. PER. SQ. IN.; R. P. M. 445. FAILS SPINDLE OIL.

No. 1	Oil alone	6 Drops per Minute	No. 3	0.35 per cent graphite	8 Drops per Minute
No. 2	" "	8 " " "	No. 4	" " "	4 " " "

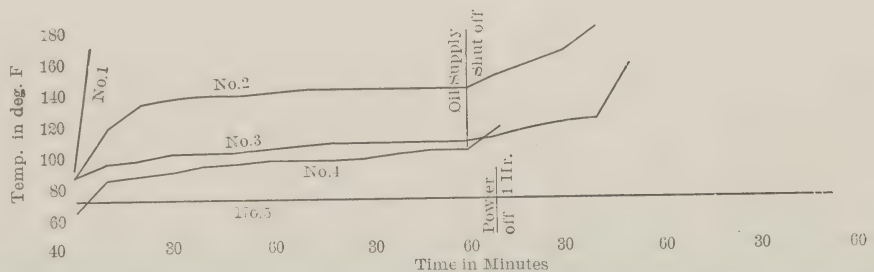


FIG. 5 TEMPERATURE CURVES FOR LUBRICANTS OF VARYING VISCOSITY, WITH AND WITHOUT GRAPHITE. PRESSURE 150 LB. PER SQ. IN.; R.P.M. 444

No. 1	Auto cylinder oil (alone)	6 Drops per Minute	No. 3	Fuel oil + 0.35 per graphite	8 Drops per Minute
No. 2	" " " + 0.35 per cent graphite	4 " " "	No. 4	Kerosene " " " "	8 " " "
			No. 5	Water " " " "	8 " " "

doubt the fuel oil possesses the quality of oiliness in a very slight degree, enabling it in the beginning of the test to take a lower coefficient than kerosene, which maintains a considerably higher coefficient for a few minutes, until the continuous film of graphite has formed and reduced the coefficient to its normal condition. It is evident that the fuel oil also possesses a certain oiliness which enables it to begin the test with a coefficient that changes only slightly during the entire period, including also an endurance test extending through two hours before the oil breaks and with only a slightly increased coefficient of friction after the oil supply was shut off. Another feature worthy of note is the comparative endurance of the three oils. While kerosene, under a bearing load of 150 lb. per sq. in., maintains an extremely low coefficient, the fact that it breaks immediately when the oil supply is shut off indicates that it has not the power to form a coherent graphite film, a power which is possessed to some extent by the fuel oil and in a marked degree by the spindle oil.

26 Fig. 4, load 150 lb. per sq. in., 445 r.p.m., gives the effect on a spindle oil of a variable feed. In one test on the oil alone the oil supply was regulated with the object of breaking the oil at the beginning of the test, and also its behavior was noted, under an oil supply that enabled it to perform its functions as a lubricant. The effect of graphite on the lubricating quality of the oil is also shown in Curve 3 and Curve 4, Curve 3 representing a feed of 8 drops per min., Curve 4 representing a feed of 4 drops per min. The diminished coefficient in Curve 4, as compared with Curve 2, represents the lubricating effect of graphite, and this effect is still further shown by the increased endurance test in Curve 4; it will also be observed that besides showing diminished friction, Curve 4 is based on an oil supply due to the graphite, one-half that of Curve 2 of the oil alone.

27 In Fig. 5 curves are shown which represent the temperatures recorded in tests of friction presented on Fig. 2 and Fig. 3. As in the previous charts the load is given as 150 lb. per sq. in. for the automobile oil, fuel oil and kerosene, and 70 lb. per sq. in. for water. The speed was 444 r.p.m. in all but the test with water, where it was 446 r.p.m. The test of the automobile oil alone showed an immediate rise in temperature, corresponding to the breaking point of the oil, which is shown in the friction test. It is interesting to compare this temperature with that of Curve 2, automobile oil and 0.35 per cent graphite, in which the temperature rises within twenty minutes to a definite point and then continues in a nearly straight line with little variation to the point where the oil supply was shut off at the end of two hours.

Curve 3, representing the temperatures of fuel oil and graphite, also shows a very slight variation after 30 min., when the stable conditions of lubrication were established. A difference in temperatures of approximately 25 deg. is shown between the curves of the automobile and fuel oils, which must represent the larger escape of energy in the form of heat from the bearing, due to the greater internal resistance of the automobile oil. The temperatures of kerosene with graphite, as shown in Curve 4, are approximately 10 deg. lower than those in the fuel oil curve, due to the still smaller internal resistance of kerosene. Bearing in mind the small difference between the specific gravity of the fuel oil, approximately 35 deg. Beaumé, and that of kerosene, approximately 45 deg. Beaumé, the difference in temperatures of these two curves is a good example of the accuracy in observation possible in these tests. Perhaps the most striking feature on this chart is the curve presenting the temperatures for water and graphite; here, as in the curve of friction for water, this curve is shown for only four hours, but the test actually extended through a period of 15 hours, during which time there were several stops in which, as shown on this chart, the temperature at the start was the same as that at the time of interruption. It will be observed that this chart shows an extremely low temperature, 65 deg., practically the same as the room temperature, which it never exceeded by more than 5 deg., and that it is essentially a straight line from start to finish. In this use of water as a vehicle for the graphite there is nothing to interfere with the best work that the graphite is capable of performing.

28 Among the various classes of lubricating oil examined in this work considerable attention has been given to the behavior of heavy engine and cylinder oils, both straight hydrocarbon oils and compounded oils. A special form of bearing was constructed, consisting of a cast-iron frame with a hollow chamber for introducing steam, and a babbitted face using the exceptionally hard babbitt previously described. In some of these tests a bronze bearing similarly constructed, but maintaining the bronze face, was employed. But in general it was observed that the results were less satisfactory with the bronze than with the babbitt bearings, in testing not only the heavy oils but the other classes of oils examined. Hard babbitt seems to possess certain peculiar qualities adapted to the various details and variations in speeds, loads, and temperatures, which are not found in the same degree in the bronze alloys. To show the results obtained in testing cylinder oils, charts are here presented on three commercial products, the American cylinder oil, Galena cylinder oil, and "600 W" cylinder

oil. Tests were also made on the influence of graphite on these oils, with reference to the frictional coefficient and endurance of the oils. The physical constants of the oils are also given for comparison, especially of specific gravity and viscosity. The general procedure of the tests included a continuous run for two hours at which time the supply of oil was shut off.

29 In Fig. 6 of the American cylinder oil, which is a straight hydrocarbon oil, the data of the tests include the use of the bronze bearing, a supply of lubricant at the rate of four drops per minute, a total pressure of 1200 lb., and a speed of 245 r.p.m. The curve of the straight oil begins at a somewhat higher coefficient that is maintained

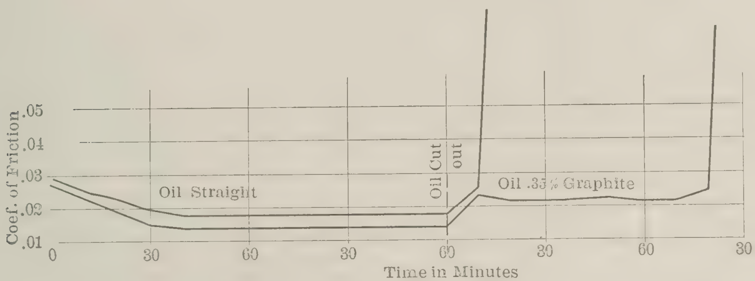


Fig. 6 American Cylinder Oil, with and without Graphite. Bzonze Bearing; 4 Drops per Minute; 1200 lb. Pressure; Temperature 210 deg. Fahr.; Viscosity 100 deg. at 212 deg. Fahr.; Flash 440 deg. Fahr.; Specific Gravity 0.961.

after the first half-hour, when normal conditions are established, and it then proceeds in a straight line with no variation to the point where the feed is stopped. The endurance run of this oil is doubtless considerably shorter than it would have been with the use of babbitt bearings; in fact this was demonstrated in another test in which babbitt was used. With graphite the oil follows closely the direction of the other curve but with a very considerable diminution in the coefficient of friction. It further appears in the endurance test that the graphite carries the load with slightly increased friction for a period of 1 hr. 20 min., which would doubtless have been considerably prolonged if babbitt had been used.

30 Fig. 7 presents results obtained in tests of the "600 W" cylinder oil, with and without graphite. A comparison of physical constants with those in Fig. 6 shows a materially lower specific gravity and somewhat higher viscosity. In these tests the same total

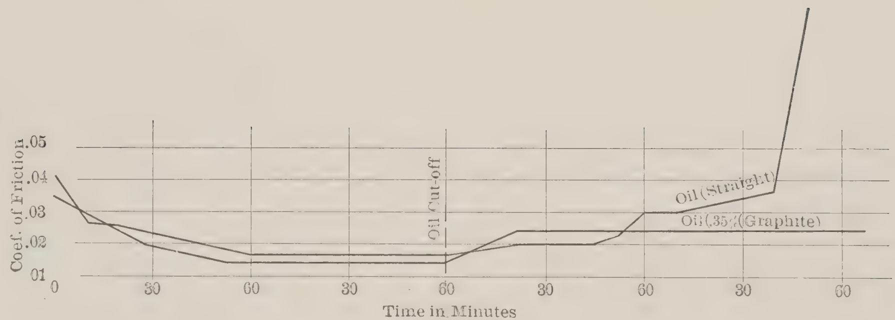


FIG. 7 "600 W" CYLINDER OIL, WITH AND WITHOUT GRAPHITE. BABBITT BEARING; 8 DROPS PER MINUTE; 1200 LB. PRESSURE; TEMPERATURE 210 DEG. FAHR.; VISCOSITY 150 AT 212 DEG. FAHR.: SPECIFIC GRAVITY 0.903; FLASH 530 DEG. FAHR.

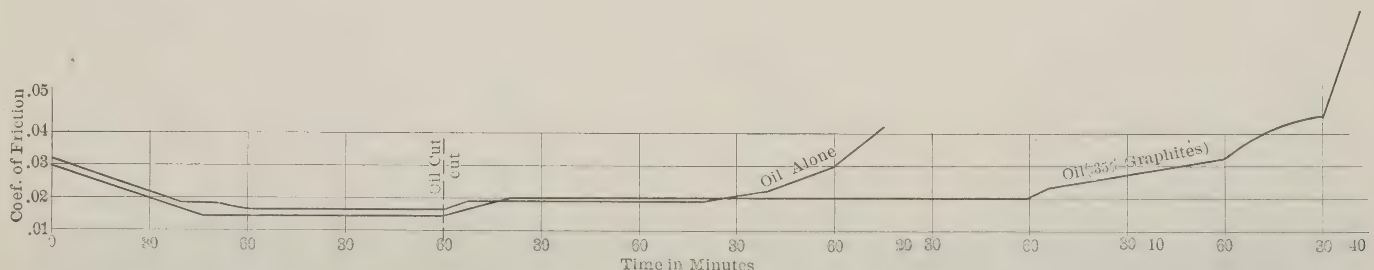


FIG. 8 GALENA CYLINDER OIL WITH AND WITHOUT GRAPHITE. BABBITT BEARING; 8 DROPS PER MINUTE; 1200 LB. PRESSURE; TEMPERATURE 210 DEG. FAHR.; VISCOSITY 116 AT 212 DEG. FAHR.; SPECIFIC GRAVITY 0.947; FLASH 266 C.

pressure, 1200 lb., and the same speed, 245 r.p.m., were used, but the oil feed was double that in the preceding tests and the babbitt bearing was employed. On account of the greater viscosity the straight oil showed at the beginning a considerably higher coefficient and the tests continued one hour before the oil had reached normal conditions, which it maintained until the feed was stopped and doubtless would have continued indefinitely. After the oil was shut off lubrication was maintained with some slight irregularity and increased friction during 1 hr. 40 min., the point at which it broke. Similar conditions are observed in the curve which expressed the variation in the coefficient of friction of this oil with 0.35 per cent graphite; it begins the test with a somewhat lower friction, reaching normal conditions sooner than the straight oil, continues in a straight line to the point where the supply is stopped, and then still continues in a straight line with somewhat increased friction. The endurance curve would doubtless have continued for a considerably longer time but the power was shut off at the point where the curve terminates. A marked influence of graphite on the behavior of this oil is plainly apparent in a comparison of these curves.

31 In applying tests to the Galena cylinder oil, with and without graphite, the same feed, load and pressure were used as with the preceding oil and the tests were made on a babbitt bearing. In viscosity this oil is somewhat less than the preceding oil, the specific gravity somewhat higher. Both curves in Fig. 8 begin with slightly lower coefficient at 0.03, and this difference is maintained until the oil is shut off and for $1\frac{1}{2}$ hours on the endurance test. To reach normal conditions the straight oil ran for 1 hr., the oil with graphite 45 min. After the feed was stopped, the curves proceed regularly with slightly increased friction, the oil alone practically breaking in $1\frac{1}{2}$ hours, the oil with graphite proceeding with perfect regularity for three hours, changing slightly during the next hour and breaking at the end of $4\frac{1}{2}$ hr. The tests represented in Figs. 6, 7 and 8 are not intended to present the comparative efficiency of these particular oils but to demonstrate the application of this method of testing and also to compare the effects of deflocculated graphite.

32 The results presented in this paper, with reference to the uses of graphite as a solid lubricant, indicate that in the deflocculated form it can readily be applied with great economic efficiency in all forms of mechanical work. One of its most characteristic effects is that of a surface-evener, by forming a veneer, equalizing the metallic depressions and projections on the surfaces of journal and bearing; and being

endowed with a certain freedom of motion under pressure, it affords the most perfect lubrication. In automobile lubrication the great efficiency of graphite, in increasing engine power, in controlling temperatures, and in decreasing wear and tear on bearings, has been brought out in a series of tests conducted by the Automobile Club of America. In connection with the reduction in friction of lubricating oils by graphite the extremely small proportion necessary is worthy of note; the proportion used in this work is equivalent to one cubic inch of graphite in three gallons of oil. The curve of temperature for Aquadag, an increase but slightly above that of the surrounding atmosphere, demonstrates an important economic quality of controlling temperatures in factory lubrication, thereby avoiding the danger of highly heated bearings, which are frequently the cause of fires.

33 In the observations described in this paper, and in fact in all the work that has been done in this field, there is not a more impressive example of the efficiency of graphite in lubrication than that presented in the curves of friction and temperature of water and graphite; for water serving merely as a vehicle and completely devoid of lubricating quality, the graphite is permitted to perform its work without aid and with no limiting conditions.

DISCUSSION

TAN BARK AS A BOILER FUEL

By David Moffat Myers, Published in The Journal for October

ABSTRACT OF PAPER

The average fuel value of spent hemlock tan is about 9500 B.t.u. per lb. of dry matter, which is about 35 per cent of its total moist weight in the fireroom. The available heat value per pound as fired is 2665 B.t.u. One ton of air-dry hemlock bark produces boiler fuel equal to 0.42 tons of 13,500 B.t.u. coal. The degree of leaching does not affect the number of heat units per pound of dry matter, but of course reduces the available material.

Boiler tests under normal conditions show thermal efficiencies of from 58 to 68 per cent, and a higher efficiency has been obtained under special conditions.

Tan presses have produced no marked increase in boiler and furnace efficiency when tested; with the same efficiency, however, an increase of about 5 per cent of steam for the same amount of tan bark may be expected owing to the increase of available heat units in the "tan as fired." Grate surfaces should be materially reduced when tan is pressed.

Mixing coal with tan under proper conditions increases both the capacity and the efficiency of boiler and furnace.

Conditions productive of best results have been: ample combustion space, and a refractory arch over the entire grate; no less than 0.5 in. and preferably of 0.6 in. water-gage draft with ample draft passages; feeding through holes in top of furnace in small quantities and at frequent intervals to approximate the rate of combustion; constant care to prevent blow-holes; a small shallow-fired furnace; a high arch above the fire (which is about the most important single requirement); proper ratio of heating surface to grate surface for local conditions; the pressing of tan under certain conditions.

ADDITION TO PAPER BY AUTHOR

The following illustrations and descriptive matter have been added by the author and should therefore be considered a part of the paper as presented at the December meeting.—Editor.

73 The following illustrations, reproduced from working drawings and sketches, will give some idea of the construction of furnaces for burning spent tan bark, sawdust, and bagasse.

74 Fig. 2 is a working drawing of the self-feeding tan-burning furnace designed by the writer and referred to in Par. 12 and Table 4.

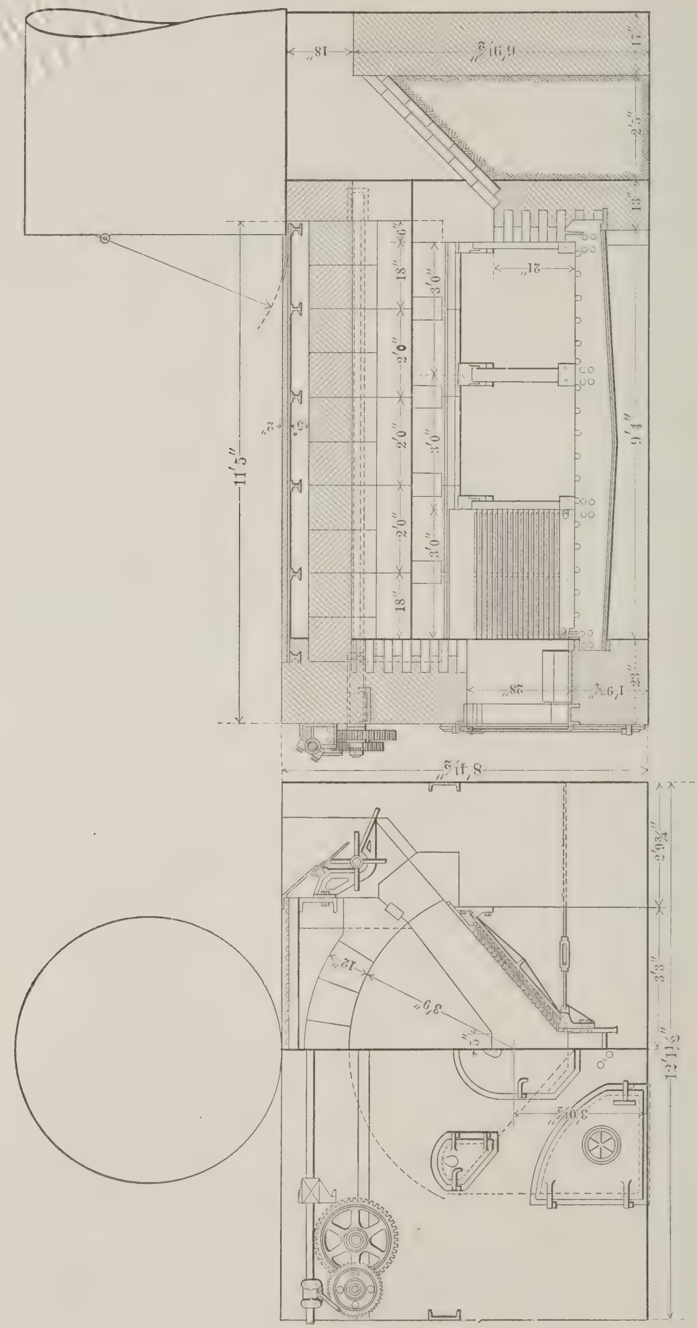


FIG. 2 END AND SIDE SECTIONAL ELEVATION OF THE MYERS TAN-BARK FURNACE

75 Fig. 3 shows the construction of the grate bars, which provide the horizontal draft opening tending to produce the draft action referred to in Par. 13.

76 Fig. 4 shows the application of these grates to bagasse burning. The stokers are done away with in this case, the fuel being fed by gravity to the feed chutes with weighted flaps which are used all over the islands of Cuba and Porto Rico. This burner has not yet been applied to bagasse burning.

77 Fig. 5 and Fig. 6 show the types of tan furnaces found by the writer in common use throughout the country. Fig. 5 shows what was known as the old Hoyt furnace. It was originally designed when tan bark was so plentiful that it was necessary to burn it. The writer has found these furnaces with inside lengths as great as 24 ft. on the grate surface. Fig. 6 shows a more modern type of burner designed to give a more even distribution of the fuel on the grates.

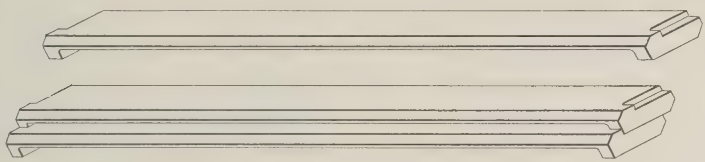


Fig. 3 Detail of the Grate Bars of the Myers Furnace

78 Fig. 7 shows a more up-to-date furnace designed for the hand firing of a mixture of coal and tan, the coal being mixed with the tan before entering the furnace, which is supplied with shaking or shaking and dumping grates. When coal is mixed with tan in any considerable proportion, more air is required for combustion, the best air spacing in the grate bar being found to be $\frac{3}{8}$ in. The percentage of draft area for this purpose should be about 40 to 50 per cent, depending upon how large a percentage of coal is used with the tan.

79 Fig. 8 shows what is known as a hump-back grate, which has been installed in different tanneries for the purpose of increasing the consumption of fuel in a given furnace. For instance, in a plant that had trouble in consuming all its tan bark, the writer merely took out the grate bars and put in a ridge bar as shown and converted the grate surface into the hump-back form. The result was that the consumption of tan bark per furnace was increased from 12 tons per day on the dry bark basis to 15 tons.

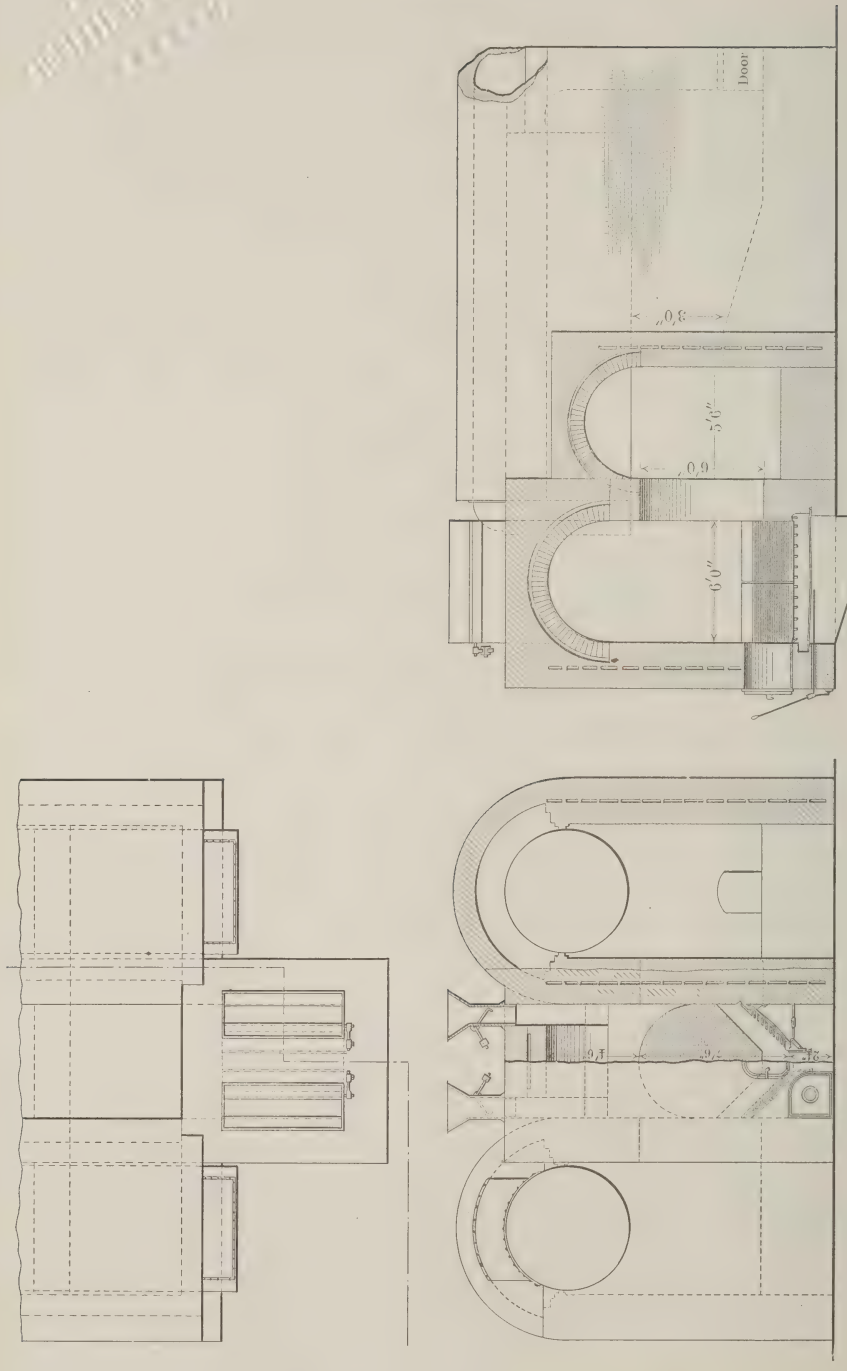


Fig. 4 Partial Plan, Side and End Elevations of the Myers Furnace for Burning Bagasse

80 Fig. 12 shows what was known as the Thompson type of tan furnace. The MacMurray furnace, with a convex grate surface and feed pipes, is a type quite a number of which the writer has seen in operation.

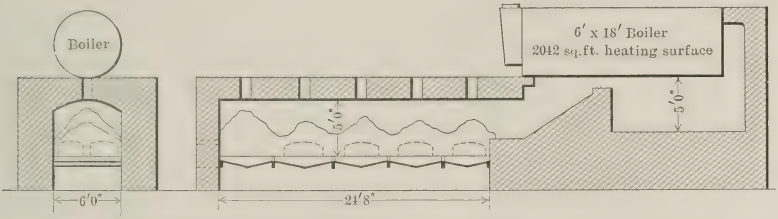


FIG. 5 THE EARLY HOYT FURNACE FOR BURNING TAN BARK

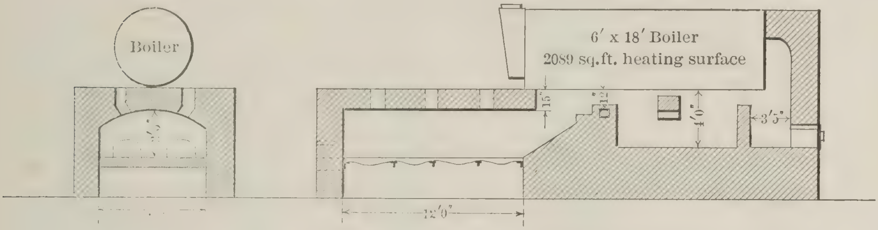


FIG. 6 A TAN FURNACE WITH SIX FEED HOLES

THE SETTING HAD AIR ADMISSION IN THE BRIDGE WALL AND A BAFFLE ARCH IN THE COMBUSTION CHAMBER. VERY GOOD RESULTS WERE OBTAINED.

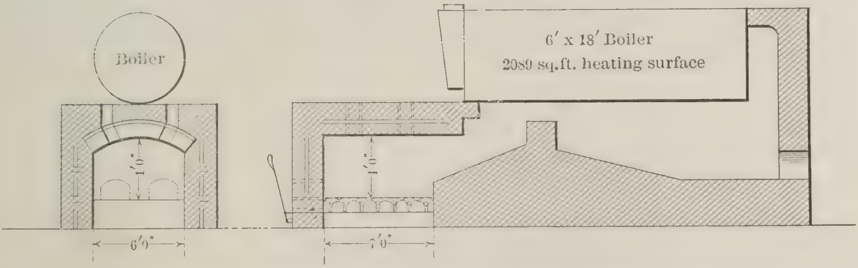


FIG. 7 A FURNACE WITH SHAKING GRATES FOR BURNING A COAL AND TAN MIXTURE

AIR SPACES OVER FIRE ARCH AND IN WALLS OF FURNACE AND BOILER WALLS. DISTANCE FROM GRATE TO TOP OF ARCH INSIDE SHOULD NOT BE LESS THAN 4 FT.

81 Fig. 10 is another form of tan furnace which gave good results in a plant in the South. The hump-back form of grate is reversed something like that used in the writer's stoker furnace, except that the tan is fed through a number of feed holes along the upper edges of these grates. This furnace was designed by the foreman in a Southern tannery.

82 Fig. 11 shows a design of the writer's for an adjustable gravity-feed furnace for burning tan or sawdust. The feed chutes are rectangular in section and contain adjustable chutes to regulate the depth of tan on the grates for any condition of draft, etc.

DISCUSSION

Albert A. Cary. The furnace described by Mr. Myers consists of an extension in front of the regular boiler setting, with a number of circular stoke holes, or openings through the top arch, over the

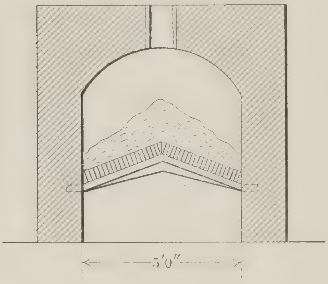


Fig. 8 Cross Section of Furnace with Hump-Back Grates and Bearing Bar

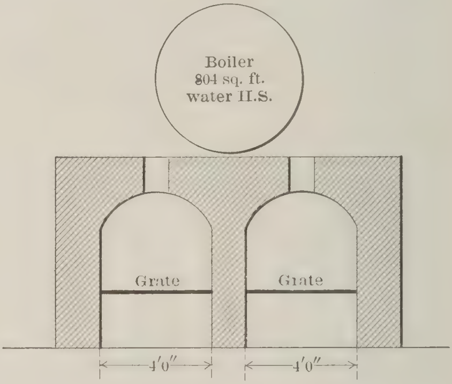


Fig. 9 Cross Section of a Double-Arch Tan Furnace of the Thompson Type on which a Test was Run

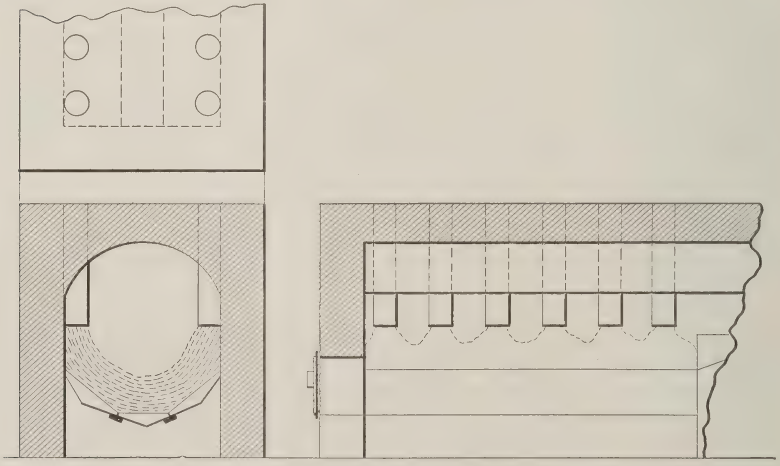


Fig. 10 Bush Tan Furnace with Multi-Tube Feed.

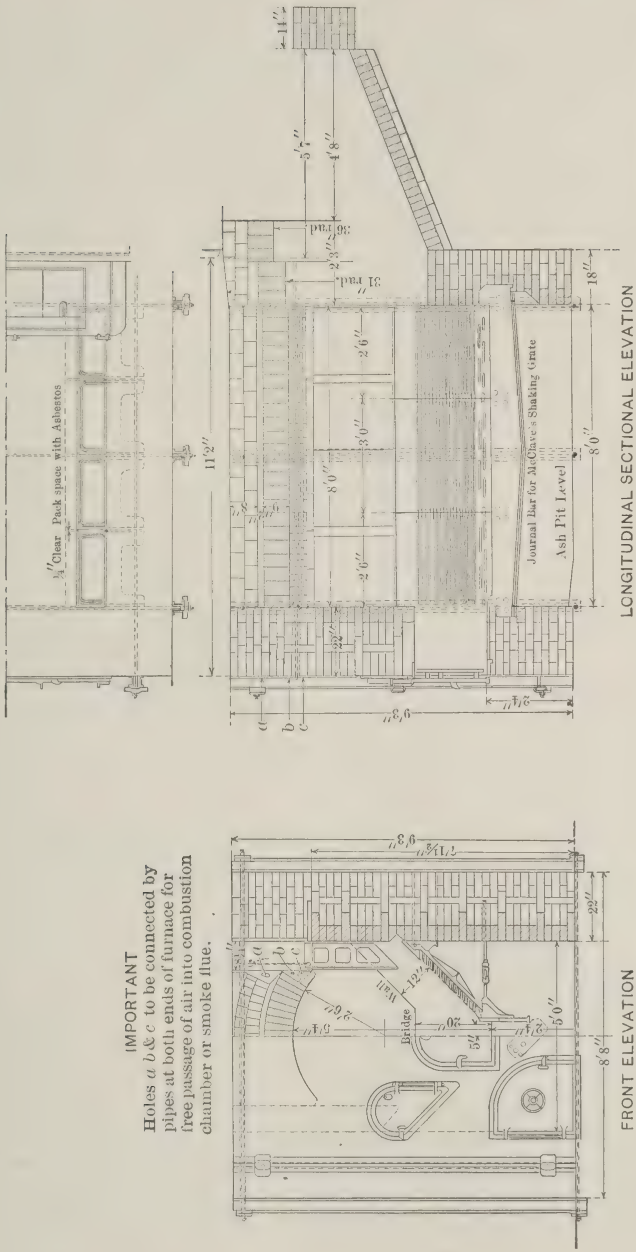


FIG. 11 A MYERS FURNACE WITH ADJUSTABLE GRAVITY FEED. ADAPTED FOR BURNING TAN BARK OR SAWDUST

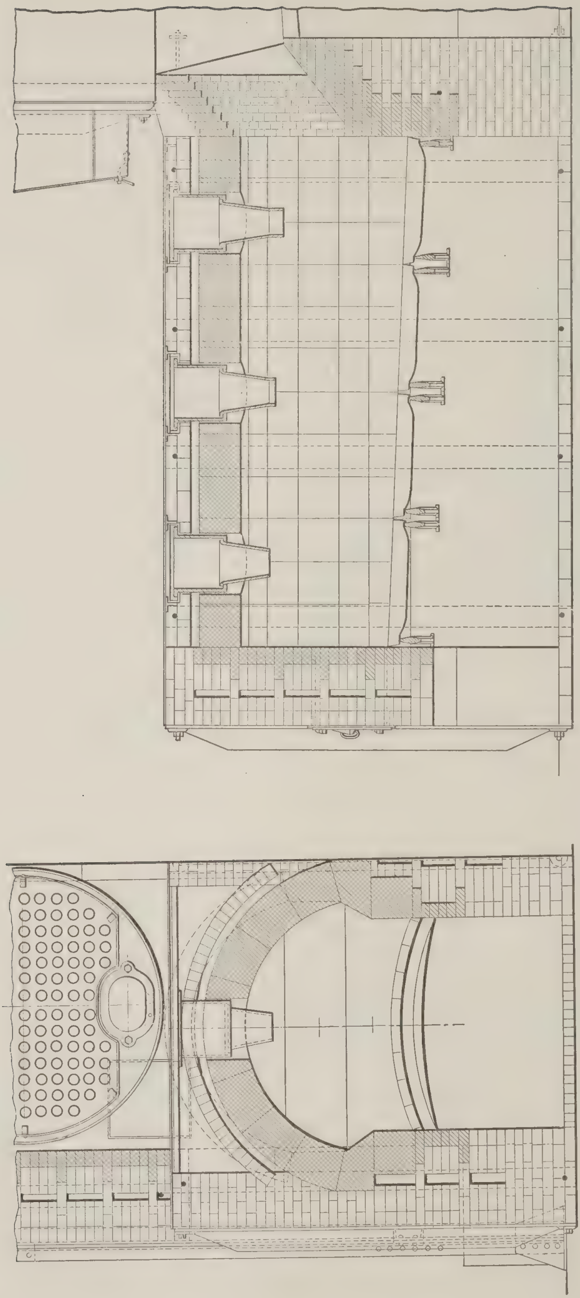


Fig. 1 The McMurray Furnace for Burning Spent Tan Bark.

grate. No little trouble has been experienced with this construction, due to the destruction of the lower end of these circular fire-brick tubes through which the fuel is charged to the furnace.

2 If these stoke holes could be always completely filled with fuel, so as to prevent inrushes of air, this destructive effect could be materially checked. However, as the method of charging fuel by hand is an intermittent one, the upper end of the cone of spent tan bark, soon after charging, drops below the level of the top of the arch, the inrushing air meets the hot furnace gases at these points and intense combustion results. For this reason, and due to the fact that when the excessive moisture in the fuel rises as a vapor against the arch, rapidly abstracting its heat (to become superheated steam), the fire brick cracks and disintegrates, finally resulting in a chipping off of the brick-work of the reverberatory arch around the lower end of the stoke holes. Repairs are therefore frequently necessary.

3 A continuous automatic feeding device, which would keep these stoke-holes constantly filled with the moist fuel, would undoubtedly do much to relieve this trouble by preventing an excessive infiltration of air at frequent intervals of time. Mr. K. McMurray of New York, has devised a very ingenous method for overcoming this trouble in hand-stoked furnaces. Fig. 1 shows both front and side sectional elevations of this furnace.

4 In the stoke hole is fitted a circular lining of cast iron which does not extend to the level of the inside of the arch. The lining is finished with a shoulder which diminishes the diameter of the opening by about two inches. A tube or open thimble drops into this frame, being held by a rim cast around its upper end. The lower end of the thimble extends about a foot into the furnace.

5 The fuel charged into the stoke hole falls through the thimble, and forms a cone-shaped pile below it on the grates. When the stoke hole becomes uncovered, the in-rushing air causes the intense combustion to take place, not on a level with the brick-work, but at a level below the thimble, and the life of the fire-brick arch at the stoke-hole openings is thus greatly prolonged. The ends of the cast-iron thimbles burn off gradually, but they cost but little, and are easily pulled out and new ones are inserted in their place.

6 Another trouble met with in this type of furnace is the rapid burning away of the fuel next to the side walls and the consequent large infiltration of air from the ash pit. This trouble has been largely overcome by reducing the width of the furnace by about 1 foot at the grate level, as shown in the front sectional elevation. The ledges

formed on either side of the lower part of the furnace support the cone of charged fuel on each side, thus keeping the grate effectually covered with fuel.

7 In this construction it will also be seen that instead of using a flat grate, the grate bars are curved so that the grate surface is higher at the center of the furnace than at the sides. This design decreases the thickness of the fuel bed under the stoke holes and causes a thickening of the fuel bed at the sides of the furnace.

8 Since water can be evaporated in the furnace itself only at a great loss, every practicable facility should be utilized for depriving the wet fuel of its moisture. Mr. Myers has mentioned the comparatively small gain from pressing the moisture out of the spent tan. I have used special rolls for extracting the water from moist fuels, with a desirable gain resulting. These rolls are of cast iron and run in pairs, one roll being about 12 in. in diameter, the other about 14 in. and both held together by heavy springs. As both rolls are revolved at the same number of revolutions per minute their surface speeds are necessarily different. The faces of the rolls are roughened by having a shallow checker work pattern cast upon them. The fuel is fed to the rolls continuously, and due to the tearing or macerating action between the rolls faces more than double the amount of water is thus worked out, as compared with the press results given in Mr. Myers' paper.

9 In one case, where the chimney was located some distance from the boilers, a wrought-iron rectangular flue was used to connect them, a shallow iron trough being formed on the surface of the flue by having the edges on the two vertical sides continued above the level of the top. The other three sides of the flue were covered in the usual way. The moist fuel was fed upon the chimney end of the flue and was drawn by a conveyor towards the boiler and over its top, whence it was delivered on top of the extension furnace. A small evaporation of moisture took place, sufficient to make this device desirable. The heat from the top of the boiler and the extension furnace may also be used in this way. The waste heat from the boiler may also be used to pre-heat the air delivered to the ash pit. I know of no condition where pre-heated air can be used to better advantage than with moist fuels.

10 Mr. Myers has spoken of the advantage of the high furnace over the low furnace. My experience thoroughly endorses this. When the moist fuel is charged into the hot furnace, a cloud of steam is evolved, which, when crowded down upon the burning fuel in

a low furnace, hinders combustion. A sufficient amount of steam would eventually extinguish the fire.

11 In addition to the effect of moisture described in Par. 7 and Par. 8, the large space occupied by the steam in the combustion chamber interferes with the combination of oxygen and the combustible gases evolved from the fuel.

12 In the flue-gas analysis obtained with moist fuels, of course the water in the gases condenses and is not accounted for in the analysis given.

William Kent. I consider this the most important paper on the subject of tan bark as a boiler fuel which has appeared in over thirty years. The only other paper that I know of is one by Prof. Thurston published in 1874 in the Journal of the Franklin Institute. He made some boiler tests on tan bark for fuel, using two different styles of furnace, some of his results being better than those given by Mr. Myers. I think that still better results are yet to be obtained from the use of tan bark as a fuel, by compressing out as much as possible of the moisture and using the waste heat of gases to dry the bark before it is put in the furnace. For burning the bark we must have a large fire-brick combustion chamber and give plenty of time to the burning of the gases, and then we will get as near the theoretically possible economy as can be expected.

2 The principal cause of poor economy in the burning of tan bark, besides the difficulty of securing good combustion in the furnace, is the amount of heat that is carried away in the shape of superheated steam in the chimney gases. If the bark, after being partly dried by compression, were further dried in a rotary drier by the waste heat from the chimney gases, there would be a very important gain in economy.

3 I have made a calculation showing the theoretical results that may be obtained in burning tan bark of different degrees of moisture under certain assumed conditions, the results of which are given herewith: The dry bark is assumed to have the following composition $C = 0.50$; $H = 0.06$; $O = 0.40$; N and ash $= 0.04$. Substituting in Dulong's formula, $14{,}600\ C + 62{,}000 \left(H - \frac{O}{8}\right)$, its heating value is 7920 B.t.u per lb. Bark containing 20 per cent moisture would have a heating value of $0.80 \times 7920 = 6336$ B.t.u.

4 Assuming the chimney gases to escape at 600 deg., the heat required to evaporate the water from 62 deg. and to superheat the

steam to 600 would be (212-62) + 970 + 0.48 (600-212) = 1306, or for 20 per cent moisture, 261 B.t.u. per pound of tan.

5 The 0.06 lb. of H in a pound of dry tan will unite with 0.06 $\times$ 8 = 0.48 O, making 0.54 lb. H_2O, which escapes as superheated steam carrying away 0.54 $\times$ 1306 = 705 B.t.u. for each pound of dry tan or 0.80 $\times$ 705 = 564 B.tu. for tan with 20 per cent moisture.

6 Assuming 25 lb. of air to be required per pound of C + H in the fuel or 25 $\times$ 0.56 = 14 lb. of dry tan, the heat carried away by this air heated to 600 deg. is 0.24 $\times$ 14 $\times$ (600-62) = 1808 B.t.u. per pound of dry tan or 1446 B.t.u. for tan with 20 per cent moisture. Using the figures thus found the following table is constructed:

Moisture	B.t.u. per lb. wet tan	Losses of heat due to Moisture	H in fuel	Heating air	Sum of losses	Net heat value B.t.u.	Efficiency per cent	Lb. Evap. per lb. wet tan
0.20	6336	261	564	1446	2271	4065	64.2	4.19
0.30	5544	392	493	1266	2151	3393	61.2	3.50
0.40	4752	522	423	1085	2030	2772	57.3	2.81
0.50	3960	653	352	904	1909	2051	51.8	2.11
0.60	3168	784	282	723	1789	1379	43.5	1.42
0.70	2376	914	211	542	1667	709	29.8	0.73
0.80	1584	1045	141	362	1548	36	2.5	0.03

7 Suppose that tan with 60 per cent moisture were dried to 20 per cent before being put into the furnace, using for this purpose the waste heat of the chimney gases, we would then have 0.40 dry tan + 0.60 moisture dried to 0.40 dry tan + 0.10 moisture, 0.50 water being removed. Suppose the moisture and the waste gases left the drying chamber at 300 degrees. Each pound of water dried out would take (212 − 62) + 970 + 0.48 (300 − 212) = 1162 B.t.u. and 0.5 lb. would take 581 B.t.u. The H in the 0.40 lb. of dry tan would make 0.216 H_2O, which would take away 0.216 $\times$ 1162 = 251 B.t.u. Heating the air would take 0.40 $\times$ 14 $\times$ 0.24 $\times$ (300 − 62) = 320 B.t.u. The sum of these is 1152, which subtracted from 3168, the total heating value of tan with 60 per cent moisture, leaves a net value of 2016 instead of 1379, the figure given in the table. The efficiency would be 2016 ÷ 3168 = 63.6 per cent, instead of 43.5 per cent, and the evaporation from and at 212 deg. 2016 ÷ 970 = 2.08 lb. instead of 1.42 lb.

Prof. F. R. Hutton. In 1874, the late Robert H. Thurston presented a paper on The Efficiency of Furnaces Burning Wet Fuel,

before the American Society of Civil Engineers.[1] At that date few engineers were paying attention to fuel economy, and there was little widespread knowledge as to the details by which it would be obtained. There was of course no formulated code for boiler testing. This paper introduced the writer at that time to the problems of boiler testing, and recorded for the first time for him the formulæ for the barrel type of steam calorimeter.

2 The two furnaces examined were designed to meet the same requirements as are assumed in Mr. Myers' paper; but the press which may be expected to expel a proportion of the water absorbed from the leaching process was not in use, and no data were given as to the proportion between the dry bark ground at the mill and the weight of wet leached fuel delivered at the fire room. The Dutch oven type of furnace was in use, consisting of a fire-brick chamber covered with a reverberatory arched roof. The fuel was fed in at the top of the oven through two holes in the length of the grate. The grate was of fire brick moulded to obtain a semi-cylindrical surface to the upper and lower surface of each bar unit, the concave side being downward towards the ash pit. A large proportion of the finer tan lumps was expected to fall through the holes in the arched bars of the grate and complete their combustion there on the ash-pit floor.

3 But it is very plain from the results of the tests that the furnaces were on very much the same plane of efficiency as those reported by Mr. Myers, since the respective results of evaporation from and at 212 deg. per pound of combustible were for the Crockett furnace 4.41 lb., for the Thompson 5.68 lb. and for the Myers' furnace 5.43, 4.71 and 4.54 lb., if equal accuracy be assumed in the old tests as compared with the new. This is open to doubt, however, as certain figures were assumed or deducted from other experiments and were criticized in the discussion of the results.

4 The present paper is especially interesting to the writer, because it represents the work of a furnace designed by Mr. Myers which seems to incorporate some eminently sound principles. I think all will agree that the three cardinal principles for the complete and smokeless combustion of a reluctant fuel involve the following:

a Time enough for access of oxygen in the air to the carbon gas from the fuel.

b Temperature enough for the rapid and complete chemical union of this oxygen with carbon and hydrogen.

[1]Trans. Amer. Soc. C. E., No. 102, Vol. III, 1874, p 290.

c Room enough for each atom of fuel gas to meet the oxygen atoms with which it is to unite.

The practical attainment of these results is made more difficult when the fuel is wet and in small particles of light weight.

5 We have the conflicting conditions of a hot fire and a slow rate of combustion, to combine with an intensity of draft which shall not be high. Mr. Myers does this by using the step-grate idea, so as to admit the necessary air horizontally between the overlapping bars, whereby the dropping of fine fuel into the ash-pit is prevented: but in addition and as a special excellence of the design, the grate is made to consist of two sections facing each other with their planes parallel to the long axis of the Dutch oven and the shell of the boiler. They are, as it were, upon the inclines of a truncated capital letter V.

6 The bark is fed by a measuring stoker cylinder, which drops a determined volume upon the whole length of the upper bar at each partial revolution, and this fall of new material displaces downward some of what has been drying and growing ready to ignite from the previous charges. At the bottom of the truncated V is a dumping grate from which the residue of ash may be released at intervals.

7 The consequence of the inclination of the two grate sections, with a horizontal inflow of the air, seems to be the same as is produced in a successful form of burner for acetylene gas. The two currents of gas and draft appear to meet in the center or in the axis of the V and an intense combustion takes place there, the heat of which reverberates downward from the arch of the oven, raises the temperature of the upper layers of fuel, and stimulates the rate of the union of combustile with oxygen. Such a furnace of course is not subject to the alternations of the "famine and feast" conditions when excess of wet fuel deadens the fire and causes a smoky and slow combustion, alternating with high heat and good flame and followed in turn with burned-out spots in the fire until fresh charges came in through the holes.

8 The system is also most effective for "bagasse," the wet juicy fibre of the sugar cane after passing through the pressing rolls. This is more difficult to stoke mechanically than the comminuted bark, but the requirements for its successful combustion are very satisfactorily met. Sawdust and scrap from wood-working shops also are burned in furnaces of this design with less danger from sparks at the stack.

9 Referring to the summaries by the author, it should be plain that the geater surface must be reduced if the tan is press-treated

(Par. 56) to remove moisture. Bulk for bulk, there are more heat units per unit of volume or of weight after a volume or weight of water has been expelled than there were when the tan was saturated and not pressed. If the fire is hot enough to dissociate the oxygen and hydrogen which compose the water, the heat for such dissociation is drawn from somewhere: doubtless from the flaming gases, where the process takes place, and of course they are cooled, and perhaps killed.

9 If such oxygen and hydrogen recombine, nothing is lost, and perhaps a mechanic-thermal advantage is reaped because the hydrogen flame is longer than the carbon flame. If for any reason such dissociated hydrogen does not get a chance to recombine from lack of temperature or time or room, there is a loss. Mr. Myers' results should serve to check the claims still advanced at intervals, that the combustion of steam-gas is a source of any great possible economy.

The Author. Mr. Cary in his discussion has described the McMurray tan furnace—one of many different types and designs now in use. All the ordinary forms of tan furnaces feed the fuel through holes in the top or arch over the grate. The number and arrangement of these feed holes vary in the different designs, but they all form a bed of fuel composed of cones of tan. For this reason they are all subject to the objection made by Mr. Cary, i. e., that the fuel burns away most rapidly around the bottom of these cones where the depth of fuel is least. The central parts of the cones offer great resistance to the draft so that active combustion takes place on only a small percentage of the entire grate surface. This necessitates large grate surfaces and large furnaces with attendant radiation losses.

2 Another objection to the cone method of feeding the fuel, especially when only a single row of feed holes is employed, is that the fire is actually divided into a number of small fires around the bottom of the cones. This multiplicity of small fires separated by heaps of wet tan of low temperature results in lowering the furnace temperature and in retarding combustion.

3 In furnaces of this type with careless firing the writer has seen fully one-half of the grate surface doing no work at all in the way of any active combustion. These ill effects are best eliminated by very frequent feeding of the tan in small amounts, so that the percentage of wet tan in the furnace at any time is very small compared to the actively burning mass. High furnace temperature is thus maintained, more grate surface is active and the rate of combustion per square foot is greatly increased. The result is less grate surface required,

smaller radiation loss due to smaller furnaces and greater ease in handling and cleaning the fires.

4 In general the greater the number of feed holes the higher will be the rate of combustion and the smaller the furnace required. Rapid firing in small amounts to equal the rate of combustion in the furnace is productive of best efficiency with any of the usual types of tan furnaces.

5 Tan presses of different makes, but all of the same type described by Mr. Cary, have been experimented with by the writer. It was found that with careful adjustment and attendance the presses would equal the performance quoted by Mr. Cary but that under tannery conditions of indifferent attendance and unskilled labor the presses do not maintain their efficiency.

6 The interference of the steam gas evolved from the fuel with the union of the combustible gases with the oxygen must be overcome by providing large combustion space, preferably over the fuel bed, by special baffles or by special draft action as in the writer's design of automatic furnace shown in Figs. 2 and 3 and referred to by Professor Hutton.

7 The chemical composition of tan is assumed by Professor Kent to be practically the same as that given from an actual analysis in the author's paper. The heating value according to Dulong's formula is 7920 B.t.u. per lb., whereas the results of a large number of tests in a bomb calorimeter by Dr. Sherman, shows the heating value of a pound of dry hemlock tan to be close to an average of 9500 B.t.u.

8 I have carefully read the record of tests on tan burning furnaces made by Prof. R. H. Thurston, and presented in a paper before the Franklin Institute in 1874. Professor Kent states that some of the results there given are higher than those determined in recent practice by the writer. The two evaporative results by Thurston are given as 4.24 lb. equivalent evaporation from and at 212 deg., in the boiler per pound of combustible for the Thompson furnace, and 3.19 lb. for the Crockett furnace. The corresponding figure obtained by the writer in his automatic furnace was 5.55 lb.; that is, over 31 per cent better than Thurston's best result.

9 The writer finds that the evaporations of 5.68 and 4.41 for the Thompson and Crockett furnaces respectively were obtained by Thurston by *adding to the evaporation in the boiler the amount of moisture in the fuel evaporated from and at* 212 *deg.* A similar addition to the writer's evaporation in the boiler of 5.55 lb. would make an evaporation of 7.75 lb. including the moisture in the fuel. The latter

figure is therefore the one to be compared to Thurston's result of 5.68 lb. On the same basis of calculation the economic result of present best practice is over 36 per cent higher than the best result recorded by Thurston.

10 Moreover the highest result in the Thurston test was obtained by a rough volumetric approximation of the weight of the fuel used. It was not weighed to the fireman as in all the author's tests. Furthermore, both the weight and temperature of the feed water were merely approximated and assumed to be correct in the Thompson furnace test; whereas these values in the author's tests were all observed and recorded in a most accurate and systematic manner.

11 The accuracy and reliability of these old tests is very much to be doubted, as Professor Hutton suggests. But even if taken at their full values it is seen that the results of present practice have exceeded the old results by over 30 per cent.

12 Actually the present results are probably even higher than this, from a comparative standpoint, for the reason that in the old days of tanning, the moisture in the tan was less than in present practice. This consideration would have given the Thurston tests a decided advantage in the shape of a greater available heat value of the fuel. Thurston gives the moisture contents of the fuel as fired as 55 and 59 per cent, whereas the moisture in the writer's automatic furnace test was 65.3 per cent.

13 This increase in moisture is due to radical changes in the process of leaching the bark. Where formerly the bark was treated with cold, or nearly cold, water it now is leached at temperatures as near the boiling point as possible, and is subjected to the leaching process two or three times as long as in the former methods. This is on account of the high price of bark nowadays, which makes it pay to leach out as much of the tannin as is practically possible. Some tanneries to-day leach their bark so thoroughly that only $\frac{1}{2}$ per cent to 1 per cent of tannin remains in the spent tan.

14 The author desires to add that all results and data given in his paper are results of actual tests made under working conditions. No assumptions or theoretical calculations are involved in the conclusions. The feed water was in every case measured by means of two tanks or barrels set above a reservoir from which a separate feed pump supplied the boiler. Feed connections were so separated that it was physically impossible to pump the water elsewhere than in the boiler being tested. All connections involving a chance for leakage were blanked off. Valves were never assumed to be tight

but were proved so during the entire test by means of an open-tee arrangement which would show any leakage.

15 The temperature of water entering as well as leaving the measuring barrels was taken at frequent regular intervals. The barrels were calibrated by weighing when filled to their overflow pipes with water at the temperature which the feed water had averaged during the test.

16 The fuel was in every case weighed in equal amounts to the fireman. A sample corresponding to each 200 lb. was taken, kept in closed receptacles and at the end of the test was mixed, and quartered down to a quart or two quart sample which was sent in sealed jars to Dr. Sherman for determination of B. t. u. and moisture. All readings and observations were obtained with like regard for accuracy of results.

17 In Par. 3 Professor Hutton also compares the best results obtained by the writer with those of Professor Thurston; but as before pointed out, the results are on a very different basis and are not comparable, unless the moisture in the fuel is also added to the equivalent evaporation obtained in the boiler. If this is done the following table gives a correct comparison:

POUNDS EQUIVALENT EVAPORATION FROM AND AT 212 DEG.

Including Water In Fuel		Excluding Water in Fuel	
Thurston Tests	*Myers Test*	*Thurston Tests*	*Myers Tests*
5.68 for Thompson furnace	7.75 for Myers furnace	4.24 Thompson furnace	5.55 for Myers furnace
4.41 for Crockett furnace	6.63 for present ordinary furnace	3.19 for Crockett furnace	4.30 for present ordinary furnace

The table shows that when compared on the same basis of efficiency the art of tan burning has been greatly improved over the old methods, both with improved and ordinary furnaces.

18 Thermal efficiency is of course the safest and most accurate basis of comparing results of various boiler and furnace settings, and the highest result yet obtained in a reliable witnessed test in tan burning was 71.1 per cent. This is based on available heat in the fuel as fired after allowance is made for evaporating the moisture in the fuel. This test, which was made on the automatically stoked furnace before referred to, showed an efficiency of boiler and furnace of 54.4 per cent, based on the total heat of the fuel.

THE DESIGN OF CURVED MACHINE MEMBERS UNDER ECCENTRIC LOAD

BY PROF. WALTER RAUTENSTRAUCH, PUBLISHED IN THE JOURNAL FOR MID-OCTOBER

ABSTRACT OF PAPER

This paper is concerned with establishing a dependable method of procedure for the design of the principal sections of curved machine members, such as hooks, punch and shear frames, and the like. The basis for this method of design is the theory of the maximum straining action in hooks devised by E. S. Andrews and Prof. Karl Pearson of London University. Experimental results are submitted in support of the theory. Comparison is made of the maximum straining action predicted by the formula due to Unwin now in common use and by the analysis due to Mr. Andrews and Professor Pearson, with that found by experiments on hooks ranging from 2 tons to 30 tons rated capacity.

DISCUSSION

PROF. GAETANO LANZA. A careful perusal of the articles of Messrs. Pearson and Andrews in Drapers' Company Research Memoirs, containing the formulæ referred to by the author, reveals no flaw in the deduction of the formula for the greatest tensile stress at the section of greatest bending moment, provided it is regarded as a formula which gives the relation between the load on the hook and the tensile stress mentioned, and provided the section of greatest bending moment remains plane.

2 To determine in all cases, however, the relation between the load corresponding to a greatest stress at the above stated section, equal to the tensile elastic limit, and the elastic limit as determined by the methods of measurement employed, would, in my opinion, require a set of tests upon a series of hooks varying in their proportions to a much greater extent than those mentioned by Prof. Rautenstrauch, in which the formula of Prof. Pearson would make the two loads cited nearly equal. An example of such a case, in which this result does not hold, is a set of hooks tested under the direction of Prof. C. E.

Fuller, which were really open links of circular form, made by bending hot and annealing square bars, the side of the square being 0.75 in., where $\rho_0 = 3$ in., and where the load at the elastic limit, as determined in a similar manner to that pursued by Prof. Rautenstrauch, was 1100 lb.

3 For these hooks we should have $\gamma_1 = 1.0074$ and $\gamma_2 = 0.00658$.

4 The greatest tensile fibre stress at the section of greatest bending moment, if computed by the ordinary formula, would be 48,600 lb. per sq. in. and, if computed by the theory of Messrs. Andrews and Pearson, would be 59,300 lb. per sq. in., whereas the tensile elastic limit of the material was 30,000 lb. per sq. in.

5 In seeking an explanation of these apparently discordant facts the following observations should be kept in mind.

> *a* In a straight beam we should naturally expect the elastic limit as determined by measuring deflections to be greater than that corresponding to a greatest fibre stress equal to the tensile elastic limit, the excess varying with the span.
>
> *b* The methods used in all the experiments cited have been practically the measurement of deflections.
>
> *c* The deflections, whether of beam or hook, cannot be determined by computation from the stresses at the section of greatest bending moment only, but depend also upon the stresses at the other sections.
>
> *d* In the hooks tested by Prof. Rautenstrauch the section of the hook is a varying one in which the stresses at sections other than that of greatest bending moment have not been examined.

Hence it seems to me that before we can consider that a complete solution of this problem has been attained, we need

> *a* A more extended series of tests which shall include a considerable number of hooks of each kind.
>
> *b* An experimental determination, both for beams and hooks, of the relations between the elastic limit as determined by deflections and the load corresponding to greatest fibre stress equal to the tensile, or compressive elastic limits, in the case of varying spans and other proportions.

Chas. R. Gabriel. The results of tests of crane hooks and the figures obtained by the old and new formulæ, to which Professor Rautenstrauch calls attention, are very important as regards crane hooks and similar members of machines. If such members are not as strong as computed by the usual formula for combined bending and tension it is none too soon for engineers to be made acquainted with the fact. This is especially so because of the fact that metal beams of solid cross section, similar to the cross section of a crane hook when subjected to simple bending, show greater strength than that due to computation, at least when subjected to a breaking test. This excess of strength is so great, especially in beams of cast iron of certain cross sections, as to justify confidence in lesser dimensions for straight beam members than those that would be prescribed by calculation based on the tension and compression moments of beam sections. Similar excess over calculation, of ultimate breaking resistance by test, exists in shafts subjected to torsion.

2 One would naturally expect to find a similar excess of strength in crane hooks when put to test, but the results to which consideration is invited show quite the reverse to be the case, and are none the less valuable because disappointing.

3 As regards machine members, such as the overhung frames of presses, punching and shearing machines, etc., the large majority of such frames require to be rigid under their working loads, to an extent that renders them perfectly safe from failure by breaking. A great many points have to be considered with respect to dies being thrown out of line by the springing apart of the upper and lower arms of the frames. A small amount of such deflection would in some cases be sufficient to cause the shearing of expensive punches by the dies, rendering them unfit for the accurate work intended. In some few other cases, such as riveting, a comparatively large amount of deflection is permissible, and in some instances the proportions of a frame may be considered with respect to safety from rupture alone.

4 The cross sections of overhung frames must of necessity differ a great deal in different machines, also the relative amount of overhang or throat, depth of gap and general form of frame, whether curved similar to a crane hook, or extending straight up and down comparatively short or long distances. Various kinds of cross section such as solid rectangular, T, H, box or combination of box and rib, all have their appropriate uses. The successful designer has at times to depart considerably from formulæ that have been in use and must combine much practical judgment and observation in his work.

Factors of safety must vary from 3 to 50 or more, and stresses accordingly.

5 It is hardly to be expected that a formula for strength of crane hooks can be immediately applicable to all the various cases of overhung machine frames, but we judge it might be applicable to small frames of solid section and short overhang. Frames having a long overhang, such as represented by Fig. 1 in the paper, would in our opinion be a more trustworthy subject for the application of the useful bending formula than frames having relatively a much shorter overhang, such as indicated by the dimensions in Fig. 4. This is because the greater the overhang the more significant becomes the simple bending moment and the less significant the direct tension in the back of the frame.

6 Referring to Fig. 4, it is noticeable that the metal in the back of the frame is very thin. In frames where rigidity is the prime consideration, we believe it is a common error of designers when using cast iron to place too little material in the back. This no doubt arises from the known high compressive resistance of cast iron, without regard to its elasticity under compression; frames being designed accordingly, with regard to resistance to breaking rather than with regard to resistance to deflection. We have known of many cases where frames could be greatly stiffened by merely taking metal from the front web and putting it on the back web.

Prof. Wm. H. Burr. Professor Rautenstrauch has added a very interesting chapter to the lirerature of this subjecc, but there is perhaps a little more to the macter than has been indicated, and it bears a good deal upon what has been said by the last speaker. Doubtless the analysis based upon Professor Pearson's paper, as an analysis, is a decided improvement upon the Unwin formula, but again there comes in the same question raised in connection with reinforced-concrete beams. This analysis, whether by Professor Pearson or Professor Unwin, is based upon what is ordinarily known as the common theory of flexure, which belongs accurately only to straight beams of very small depth in comparison with the length.

2 Hooks and all such members as those shown by the author are exceedingly shorc as beams, and they are also curved. These conditions completely demoralize the analysis as based on the common theory of flexure, and it is not a matter of surprise that hooks should show so much greater carrying power than the computations would indicate. In fact, it is precisely in line with what we find in other short beams.

3 The pins at the panel points of pin-connected bridges are designed by the common theory of flexure. Yet if one should compute the extreme fibre stresses in those pins at some panel points as they have existed, they would be found to run up not only to 142,000 lb. per sq. in., but to 180,000 or 190,000 lb. in structural steel. A partial explanation lies in the fact that an analysis is used, which, strictly speaking, does not apply to these conditions. The hook and all such members, as well as bridge pins, are short, thick beams to which the usual theory of bending does not strictly apply.

4 Again, one will find that in bridge specifications, the regular working fibre stresses in pins are permitted to be at least 50 per cent greater than in the tension members of the truss; that is, one may have a working stress of perhaps 14,000 lb. in bars, and a fibre stress in tension of 18,000 or 20,000 lb., sometimes even 24,000 lb. in pins. This is due to a fact I have already mentioned, that as a matter of accurate analysis, the common theory of flexure should not be used in connection with such members; but there is nothing else to be done.

5 That again brings me back to the same point made in connection with concrete beams. The proper procedure is to settle upon some sensible working formula, just as we do in connection with the pins in bridges, make tests of such members, and deduce from these tests such empirical quantities as may be properly used in the formula, so as to make the results of the analysis in that way conform to safe and sensible practice.

George R. Henderson. That we get a rather greater strength than would be expected by the Unwin formula, especially in the case of hooks, agrees with my practical experience. A few years ago we purchased some 60-ton cranes, and when it came to the detail of the hook to lift the 60 tons, the design submitted by the manufacturers was for a hook smaller than we thought would be good practice to accept. We calculated to reduce the total strain due to the vertical stress and the bending moment to about 12,000 lb., which we considered would give a factor of safety of five with the material used. It was pointed out that the hook did not conform to the specifications, and that a larger hook was desired. These larger hooks were provided and they looked gigantic.

2 A little later the question came up again, when the manufacturers stood on their dignity and claimed that the hook was stronger than my calculations showed, and to confirm their case referred to

tests at the Watertown Arsenal, which we all consider pretty good authority. The hook tested was rated as a 20-ton hook, but it had been subjected to a weight of 162,000 lb., at which it merely bent but did not break.

3 These tests were to determine the ultimate strength, whereas the paper deals with the elastic limit; but practically, I think, the ultimate strength interests us as much as the elastic limit. By the regular Unwin formula, which has been somewhat condemned this evening, the stress per square inch in the hook, when weighted to 162,000 lb., at which it simply opened, would indicate 142,000 lb. per sq. in. fibre stress, which, of course, is absurd. So, from the actual tests, it is very evident that the hooks are considerably stronger than the Unwin formula could indicate. In discussing this matter with well known machinery builders, such as William Sellers & Company, we found that while the strain on the hooks might figure at 17,000 lb. per sq. in., from the formula, and show a factor of safety of only three, actually the factor of safety must have been five or six.

4 If possible, I would like to know how the author can reconcile these facts, compared with the practical ultimate strength tests, in connection with the elastic limit.

A. L. Campbell.[1] Table 2 of Professor Rautenstrauch's contribution shows an excellent agreement between actual test conditions and the results obtained by the formula which is the basis of his discussion.

2 A much simpler formula is used by the writer for similar computations. A crane hook or the frame for a punch is really a tension member with an exaggerated eccentric load. The maximum unit stress in such a tension member may be proved equal to

$$f_t = \frac{W}{A}\left(1 + \frac{l\,(1 - e)}{R^2}\right)$$

using the author's notations. The radius of gyration, R, is equal to $\sqrt{\frac{I}{A}}$. Applying this formula to the frame shown in Fig. 4 gives $f_t = 7600$ lb. per sq. in. This stress is 90 per cent of that given by the more complex formula.

[1]The Solvay Process Co., Detroit, Mich.

Frank I. Ellis. While the paper, together with the article in the American Machinist to which it refers, covers very fully the design of hooks, giving results which agree remarkably with actual tests, its application to shear housings is not quite clear to us.

2 We note primarily, that in the derivation of his formula the writer has assumed the entire area to be in tension i. e., the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks, it will seldom be encountered in shear housings, but still it appears to have important bearing on the form of the equations.

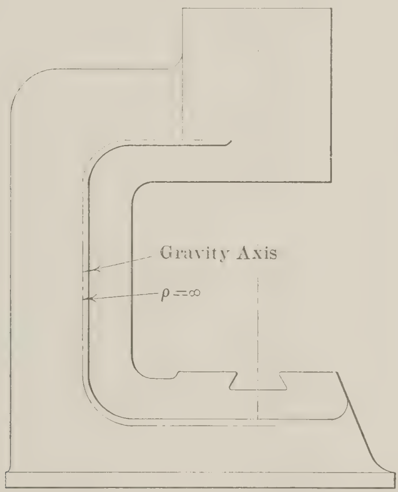


Fig. 1 Frame with Infinite Radius

3 Another point which is not quite clear to us, but is a matter of great importance, is the assumption of the value of ρ, the radius of curvature of the gravity axis of the section. In the case of a hook, this of course is quite obvious, but in machine members, such as shear housings, this seems far from being the case. For instance, in a housing of the general form of sketch shown in Fig. 1 herewith, we would have an infinite value of ρ. This would reduce the formula to a case of simple tension, which is obviously incorrect, giving stresses that would be very much less than would be obtained by actual test. On the other hand, if we consider an extreme case as per Fig. 2, where the value of ρ is very small, the stress as calculated by the formula

would be very much in excess of what could possibly exist in the actual casting.

4 The example of a shear housing which Professor Rautenstrauch has chosen as an illustration appears to us to be at variance with our experience. The stress calculated by the new formula is almost three times that obtained by the usual methods of computation. In our experience, cast iron shear housings in which the calculated

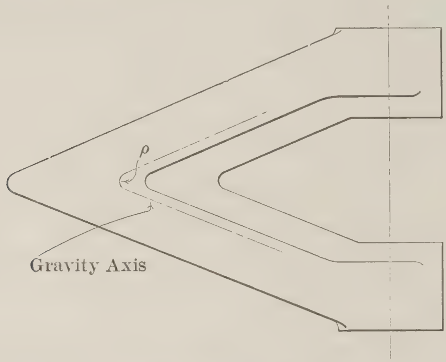


Fig. 2 Frame with Small Radius

stress is 3,000 lb. per sq. in., never break except through defects in the casting, a condition which could hardly exist if the actual stress were in the vicinity of 9,000 lb.

5 In conclusion we may say, that aside from the seeming obscurity of the paper on the above points, we consider the formula to be of considerable value in the cases it is designed to cover. We regret that we have been unable to give it the time it deserves, and trust the above points will be made clear by the author.

E. J. Loring.[1] The results of the author show such striking discrepancies from the results by the usual methods of calculation that his analysis of the problem merits the most careful consideration. These results clearly show that the stresses and particularly the maximum stress in a curved piece under the combined direct and bending load to which such hooks and gap frames are subjected cannot properly be deduced from the simple combination of direct and bending stresses as determined by ordinary analysis from the stresses in a single plane, but may be influenced to a greater extent by con-

[1] Loring Speed Gauge Co., 76 Highland Ave., Somerville, Mass.

ditions outside of the section plane, such as the relations connecting that plane with those nearby on either side.

2 This difference between straight and curved members arises from a different distribution of stress due to the variation of length of fibres at different parts of the section as taken between similar adjacent sections.

3 The usual deduction for stress in straight members commonly applied to this problem assumes

- *a* Planes remain planes after bending.
- *b* Strain is proportional to the distance from the neutral axis.
- *c* Stress is proportional to strain and therefore the stress is proportional to the distance from the neutral axis.

4 The assumption that stress is proportional to strain is true only as referring to unit strain, as long fibres will yield more under a given stress than shorter ones. In the case of straight members the adjacent minimum sections are parallel and the elementary fibres therefore all of equal length, and the assumption may be applied. In the case of a curved member, which I would define as one in which the locus of the centers of gravity of the minimum cross-sections is a curved line, these sections are not parallel, but radiate from a center of curvature so that the fibres are not of the same length throughout the section, and a correction must be made for the variation of the length of fibre before this assumption can be applied. This point has been generally overlooked or considered negligible, and in this point is to be found the explanation of the difference in results. I might add that this exemplifies the danger of applying a formula to conditions which it was not intended to represent.

5 I am not certain that I can agree with the author in the use of the theory of lateral contraction in the analysis. I cannot at this moment see why it is any more necessary in the case of the hooks tested than, for example, in the case of the test bars from which he deduced the fibre stresses. Taking only the common assumptions, with the correction for the length of fibre, as above noted, it is possible to obtain results in very close agreement with those given by the formula recommended by Professor Rautenstrauch. In place of the usual straight-line diagram of stress on the section these assumptions give the stress at any point as varying according to

$$\frac{y}{1+\frac{y}{\rho}} \text{ or } \frac{y\rho}{\rho+y}$$

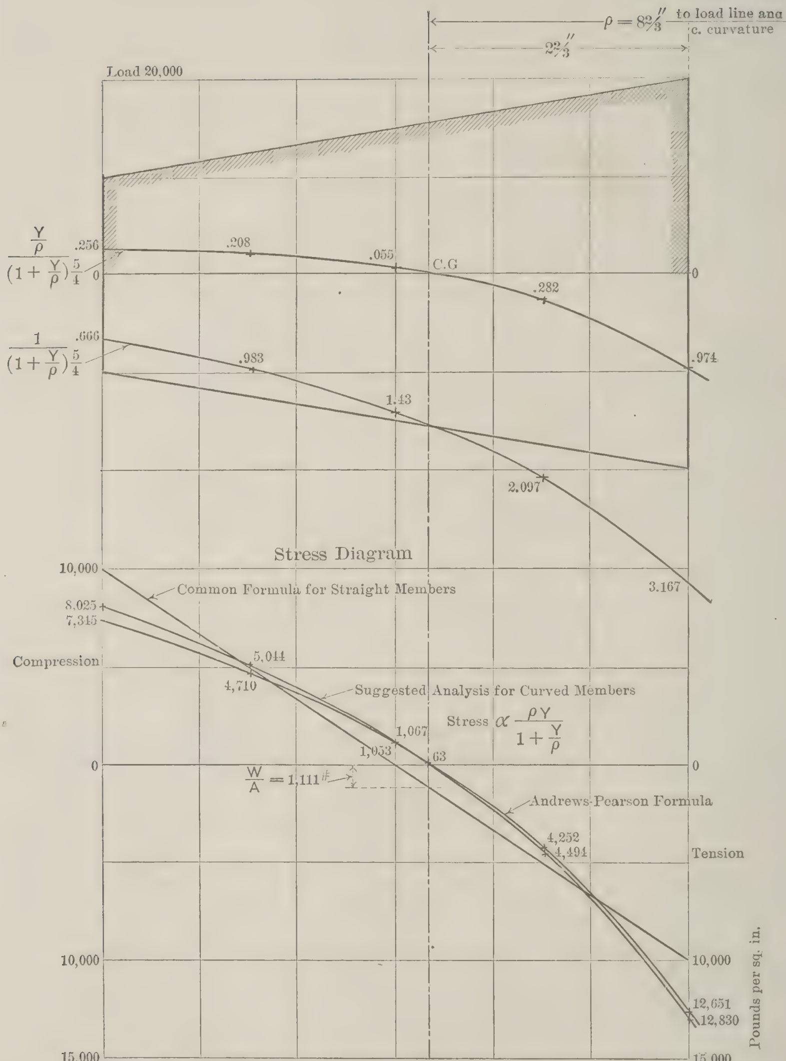


Fig. 1 Diagrams for Trapezoidal Sections of Straight and Curved Members for Equal Intensity of Stress

using the symbols of the paper, and from this may be determined this important fact; that for the case represented by the hooks, where the line of application of the load contains the center of curvature, the neutral axis contains the center of gravity. In other words, instead of the stress at the gravity axis being equal to the distributed stress as is true for straight members, the stress at this point in a member with this degree of curvature is zero, and this represents the manner in which the stress "piles up" toward the inner edge of a curved member. This condition of stress at the center of gravity would be represented in the analysis of the paper by the condition $\gamma_1 = 1 + \gamma_1$. The empirical formulæ recommended give $\gamma_1 = 1 + 1.1\ \gamma_2$ and the data on the hooks give a variation from $\gamma_1 = 1 + 1.17\ \gamma_2$ to $\gamma_1 = 1 + 0.88\ \gamma_2$ with an average of $\gamma_1 = 1 + 1.015\ \gamma_2$ so that it will be seen that this is approximately true by Professor Rautenstrauch's analysis; that the gravity axis is the neutral axis for this degree of curvature just as it is for transversely loaded beams.

6 I have applied the variation of stress

$$\frac{y}{1 + \frac{y}{\rho}}$$

given above to the solution of an assumed section and find that the stresses and their manner of variation are substantially identical, for this particular case at least, with those given by the author's method. I believe that an analysis can be made along this line that will give results very close to those shown and be more generally workable. The differential expressions for the net stress on the section and the moment of the stress are similar to those for a beam with the addition of a factor

$$\frac{1}{1 + \frac{y}{\rho}}$$

$$W \text{ varies as } \Sigma \frac{y\,\mathrm{d}A}{1 + \frac{y}{\rho}} \qquad Wl \text{ varies as } \Sigma \frac{y^2\,\mathrm{d}A}{1 + \frac{y}{\rho}}$$

It may perhaps be possible to deduce some general expression to be used as a factor of correction for curvature to be used with the usual methods.

7 The effect of the curvature is less, the greater the ratio of radius of curvature to the depth of section. In the case of hooks this means greater strength where the contour of the inner edge is elliptical instead of circular, so that the curvature at the most strained section is less. As the curvature tends to "pile up" the stress toward the inner edge, greater strength may be had by giving the hook a closer approximation to a Tee section, by which means the metal is massed better where the stresses are abnormally high. It would also appear that a high gap is stronger than a low one for the same depth, since a lesser degree of curvature is possible.

8 I must disagree with the statements in Par. 2 except as limited to curved members; also with the statement in Par. 8 that γ_1 and γ_2 are constants for all sections of similar form, except it be modified to say "of similar form, curvature and load distance." I question the significance of the quantity k in Par. 11, where it is stated to be the radius of gyration, as the value of k given for the punch frame following is the radius of gyration squared, or $\frac{I}{A}$.

9 In determining the maximum stress by the method which the author has proposed, the function γ_2 is the most important factor, and this function is obtained from the difference in area of two derived curves; the difference is small and the less the difference the greater the maximum stress. It would seem that there is great opportunity for inaccuracy in determining this factor. It also appears to me to be simpler to take

$$\gamma_2 = \frac{-\Sigma\left(\dfrac{1}{1+\dfrac{y}{\rho}}\right)^{\frac{5}{4}} \dfrac{y}{\rho}\,\mathrm{d}A}{A}$$

as originally stated for the purpose of the computation, rather than to use the value derived from it, of

$$\gamma_2 = \gamma_1 - \frac{\Sigma\left(1+\dfrac{y}{\rho}\right)^{\frac{1}{4}} \mathrm{d}A}{A}$$

for the reason that having the quantities for the determination of γ_1 for various points of the section, that is, the values of $\left(1+\dfrac{y}{\rho}\right)^{\frac{5}{4}}$

it will be simpler merely to multiply these by the respective distances from the gravity axis, plot the curve and integrate for the net area, rather than to proceed by raising the denominator to a new power and passing through all the processes anew.

10 It is stated that the standard section selected for the computation of constants for the empirical formula is not the most economic from the standpoint of equal tension and compression stresses. This is true even if the member is straight, in which case, considering the trapezoid only and omitting the curved ends, the maximum stress in compression is 85 per cent of the maximum stress in tension. All other parts remaining the same, for equal intensities of stress in the edges for a straight member, the half width of the narrow edge should be 0.095 r, as may be very readily demonstrated. The geometrical relations for the correct proportions of a trapezoidal section for equal intensity of stress in a straight member are so exceedingly simple that I want to give them here, particularly since so far as I know they have never been published. This relation is that the sides extended intersect at a distance from the far or narrow edge equal to the distance of the load line from the near or wide edge, and for the solution of this case we have

$$d = \sqrt[3]{6\,ky\,\frac{F}{f}}$$

where d = depth of section
y = distance of load line from the near edge
F = load
f = maximum stress at the near edge or far edge (equal)
and k is a design constant = ratio of depth of section to width of far edge.

11 For the case of equal stresses in a curved member of trapezoidal section with center of curvature on the load line, a similar relation may be deduced from the analysis that I have here suggested, but is not quite so simple: the point of intersection of the sides is given by the following construction. Lay off on the axis of symmetry and toward the far edge a distance from the *near* edge equal to the distance from the near edge to the center of curvature and load line. If this distance is greater than the depth of section, equal stresses may be had. If this distance is equal to the depth of section; i. e., if the point thus laid off is on the far edge, equal stresses require a triangle with this point as the apex. If the point is beyond the far edge,

divide the distance to that edge in thirds; then the stresses are equal when the sides extended intersect at the nearer point of division, one third of this distance from the far edge.

12 It will be noticed that this construction gives the radius of curvature for this limiting case[1] equal to 1.33 times the depth of section[1] instead of 1.75 as given by Professor Pearson. I have investigated this case for both degrees of curvature by the method involving the lateral contraction and find that using the formulæ given by Professor Rautenstrauch the curvature of 1.33 times the depth, measured to the gravity axis, gives a stress on the inner edge of 1.091 times that on the far edge. A similar operation for curvature of 1.75 as recommended by Professor Pearson, by his own method gives by my computations a ratio of stress of 0.912. A sharp triangular section such as this is however of little or no importance in actual construction, and the method of determining the proportions which I have given will, I think, be found to be of much more general application. I am unable to state at the present time whether a section having equal intensity of stress on the two edges is or is not the most economical of material; but presumably it is.[2]

Prof. C. E. Houghton. The agreement between the elastic limit as calculated by the proposed formula and that as derived from the tests, is to say the least, wonderfully close, and the wide variation between the experimental values and those calculated by the use of a theory that has been in common use for many years leads one to ask "Why are there not more failures in crane hooks?"

2 Objection has been made to the tests because the hooks were not loaded beyond the elastic limit. This seems to the writer to be a mistake. What the engineer is mostly interested in is the effect of loads that produce stresses within the elastic limit, since the great majority of the formulæ used for the calculation of stresses are based on theory that no longer holds true after the elastic limit has been exceeded.

3 Professor Burr has pointed out that the simple theory of flexure does not apply to curved members and Mr. Gabriel notes that stiffness and not strength is the controlling factor in many of the open-side machine frames. May not the fact that cast iron is used in the majority of such frames be another reason why the flexure formulæ cannot be expected to give correct results? The well-known fact

[1]Or depth of section equal to gap depth.

[2]Since writing the foregoing, Mr. Loring has found that the method suggested by him for the determination of the stresses—or a very similar one —is given in some detail in Hütte, from some German source dating 1902.

that the physical properties of any cast iron vary with the rate of cooling, and that the tensile strength and modulus of elasticity are not constant at all depths from the surface of any cast iron member, but vary throughout any given section, leads one to ask "Is it not more reasonable to use the simpler formulæ in the calculations for strength and to provide against possible errors by that useful and elastic term—the factor of safety?"

H. GANSSLEN.[1] The author's tests prove the correctness of Andrews and Pearson's new formula for figuring crane and coupling hooks. All the experimenters, however, seem to have limited themselves to these hooks, for which the formula appears to have been gotten out. Hook's law of the direct proportionality between stresses and strains also underlies the new formula and the fact that this law holds practically good on wrought iron, steel and similar materials would to some extent explain the good agreement of the results of tests and calculations by means of the new formula.

2 The author points out that the formula is applicable to punch and riveter frames. To generalize thus I believe is hardly wise at present, as all the various formulæ for figuring curved beams are more or less empirical and each of them is naturally proved to be true for a certain limited field of calculations only. Hook's law does not hold true for copper, cast iron, bronze, stones artificial and natural, etc., and this law giving the modulus of elasticity as constant is the basis of the formula.

3 Engineers know that the old formula for figuring a curved member in the same way as a straight beam gives too small factors of safety, but that we are now under-estimating the stresses in the throat of punch press frames $8500 \div 2450 = 3\frac{1}{2}$ times is surely saying much.

4 However, there is no use disputing the new formula in so far as tests have verified it and it is to be hoped that the author will have the opportunity of entering other fields of research besides that of crane hooks, and that of press frames would be a desirable one.

5 I have not come across a case where a punch press frame figured in the usual, but wrong way could have been $3\frac{1}{2}$ times under-estimated, roughly considered, by comparing the pressure exerted with the general behavior of the frame.

6 The old theory of flexure as applied to and compared with tests of cast iron has shown its inapplicability and this should make us all

[1] Mechanical Engineer, 404 Fisher Building, Chicago.

the more cautious in adopting the new formula for cast iron press frames before having the results of tests on hand that would justify us in so doing.

John S. Myers.[1] The author's presentation on the design of curved machine members and his article in the American Machinist of October 7, 1909 dealing exclusively with crane hooks, seem to indicate that the new theory is applicable to punch and riveter frames of the type shown in Fig. 1, where the throat is semi-circular, being struck with a radius having its center at O. In order, however, to find the radius of curvature of the gravity axis of the principal section it would seem necessary to plot points such as A, B, C, D, E, draw a curve through them, then, by trial, find the center O' of a circular arc which will pass through C and most nearly fit the curve for points intermediate between B and D.

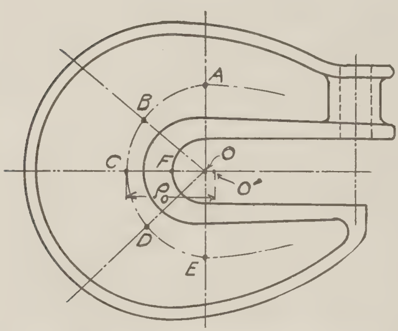


Fig. 1 Frame with Semi-Circular Throat

Curve $ABCD$ represents the gravity axis of the section. Point O is the center of the throat radius. Point O' is the center of a circular arc which approximately coincides with the gravity-axis curve for points between B and D.

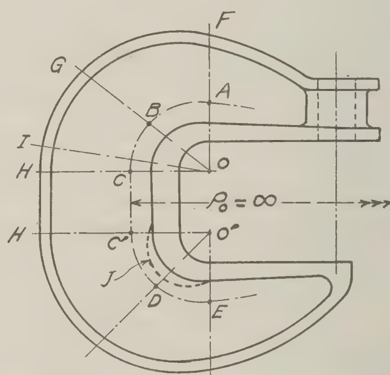


Fig. 2 Frame with Wider Gap than Fig 1

Curve $ABCC'DE$ represents the gravity axis. Between points C and C' this curve becomes a straight line; hence $\rho_0 = \infty$

2 If the above is consistent with the assumptions upon which the theory is based, it will be seen that the point O' is not necessarily coincident with O, and that to find the value of ρ_0 a layout must be made and the gravity axis of several sections determined. It

[1] John S. Myers, 2456 Almond St., Philadelphia.

is also seen that ρ_0 is not strictly a function of the throat radius nor is it equal to $OF + CF$ as one would at first suppose. This adds more complication to the problem, which is already vexatious.

3 Again, such frames are not always made with the throat struck with a single radius; in fact, this is the exception rather than the rule for quite a large class of machines, which have a wider "gap" to accommodate the work and are more like that shown in Fig. 2. Here the curve representing the gravity axis is a straight line between points C and C', in consequence of which $\rho_0 = O, O$ and it would therefore seem that the new theory did not apply to this portion of the frame. Now, if this be the case, and we design that portion of the

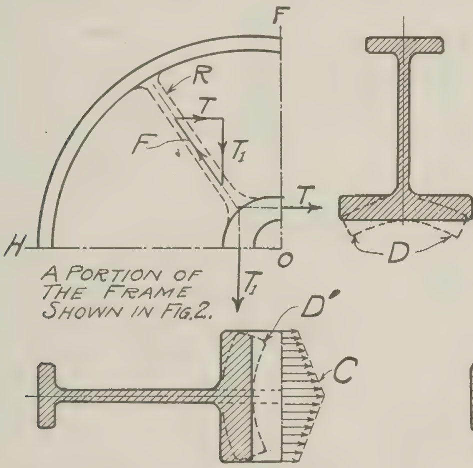


Fig. 3

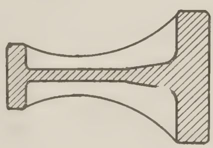


Fig. 3a

Fig. 3 Showing How the Rapid Transition of Stresses Induces Local Stresses. Fig 3a. Proposed Section

frame between OH and $O'H'$ according to the old theory of straight beams, but design section OI according to the theory of curved beams under discussion, it would appear from an inspection of the results given by Professor Rautenstrauch that section OI should have about three or four times the flange area of section OH. Of course the metal at the corners could be thickened, as indicated by the dotted line at J, but it would be out of the question to double or treble the usual flange thickness, which is what the new theory seems to indicate as necessary.

4 It would be very interesting to know how the new theory could be properly applied in such a case; whether, for instance, it is entirely applicable at the section OG but gradually merges into the old theory

at sections OF and OH; or whether it has not, as yet, been sufficiently developed to be generally applicable to sections other than those at right angles to the line of action of the force.

5 Generally speaking, a structural engineer never puts in curved tension or compression members because he knows that force either travels in straight lines or else produces bending strains; but the average designer of machinery seems to delight in curved ribs, bent levers, and the like. The average mechanical draftsman makes layouts as if he held the opinion that force travels along a curved rib in a manner somewhat similar to water flowing in a pipe and that it

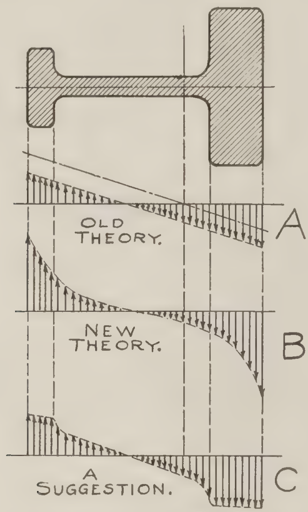


Fig. 4 Distribution of Stresses Under Different Theories

will, therefore, follow any devious or sinuous course in which he may choose to distribute the metals. Most C-frames seem to be designed on the foregoing assumption and, while it is an exceedingly difficult piece of mental gymnastics to follow the mathematics of the new theory, it is, however, quite easy to see that there are stresses induced in curved ribs which are usually ignored.

6 To illustrate the foregoing, Fig. 3 shows that portion of the frame of Fig. 2 which lies between lines OF and OH. Now, let T and T_1 represent the total tensions in the flanges on sections OF and OH respectively. By combining T and T_1 graphically it is seen that a resultant force F must, in some manner, be supplied to establish equilibrium. The most direct way of supplying such a force is by

the addition of a rib as indicated by the dotted lines at R which will distribute part of F into the web and deliver part of the force at the compression flange where there is a smaller, opposing resultant force. In the absence of any such rib the necessary force must be supplied by the web, partly through a local bending and distortion of the flanges as indicated by the dotted lines at D and D' and partly by a concentration of stresses towards the central portion of the flanges as indicated at C, this concentration being a direct result of the deformation at D'.

7 In supplying a rib R, if it was intended to carry the entire force F it would be necessary to make it about $1\frac{3}{8}$ times the average thickness of the flanges, but since the web can readily take half, or more than half, of the load it would seem that a rib of $\frac{5}{8}$ or $\frac{1}{2}$ of the flange thickness, narrowed down at the center as shown in Fig. 3a would be entirely sufficient, especially if the web be judiciously thickened and liberal fillets used.

8 As I understand the new theory it does not recognize any such concentration of stresses as indicated at C in Fig. 3 but, on the contrary, assumes a more rapid concentration towards the extreme fibres in a manner somewhat similar to that shown at B in Fig. 4. Now in view of the close accord between the new theory and the results of Professor Rautenstrauch's experiments, I am quite ready to believe that diagram A represents quite closely the actual conditions for straight beams of solid section, and that diagram B represents the most plausible theory for curved beams of solid section; but that for beams composed of heavy flanges and a light web the probable distribution of stresses is more nearly like that suggested by diagram C, and that so far as the curved form of the beam is concerned, it is not the curve of the neutral axis we are interested in but the curve of the flanges, and that this results in local bending and concentration of the stresses as already pointed out.

9 I have no well formulated theory to advance in explanation of my belief in a distribution of stresses like that indicated by diagram C but have sufficient faith in it to calculate sections of this nature by the very simple process of considering the stress to be uniformly distributed over the flange area and entirely neglecting the web; then at points where there is rapid transition of stresses, supplying ribs, thickening up the web and allowing a lower flange stress and liberal fillets. This procedure may sound crude to a scientific man, but it has, at least, ease of application in its favor and may yet be shown to be actually more scientific than the more laborious methods

usually pursued. As yet, I have not had the temerity to apply this method to large work but would like to have the opinion of those who have had experience along these lines.

The discussion concluded with an interesting talk by Carl G. Barth illustrated by a blueprint and blackboard sketches. Mr. Barth has not been able to prepare this for publication. Editor.

The Author. The test reported by Professor Lanza is interesting, but I do not feel justified in replying without a review of the entire data on the experiment. The point made by him in Par. 5 in regard to deflections, is somewhat misleading. I did not propose in my experiments to determine the relation of total deflections to the maximum stress in the hook, but rather to find the load at which the total deflection ceased to follow the straight-line law. Since the total deflection is dependent on the deflection of all the sections, it is rational to suppose that when any variations occur they are due to the fact that the "fibres" in the most strained section have been stressed beyond the elastic limit. This is all we wish to know. The most strained section is without doubt the main horizontal section. The examination of the bending moments in other sections is of no value in these determinations.

2 Referring to Mr. Gabriel's remarks: I regret that so many designers persist in applying the formulae for determining maximum intensity of stress beyond their limits of application. No computations can be made to determine *ultimate breaking strength* and I see no reason why anyone should be surprised that there is a disagreement between the "results of computations" and the results of test. I did not choose to consider the matter of rigidity, which the title of the paper would lead one to believe should be included. Rigidity is, of course, a controlling factor in die work. The dimension of the metal in the back of the frame shown in Fig. 4 of the paper should be $1\frac{1}{4}$ in.

3 Mr. Henderson's remarks that his practical experience with hooks leads him to believe that a rather greater strength exists than can be expected from the Unwin formula, qualified by his report of certain tests, would lead one to believe that he has made use of Unwin's formula outside of its field of application. Unwin's formula indicates nothing beyond the elastic limit. There exists no method of analysis which enables us to determine the relation between the load on the hook and the resulting maximum intensity of stress

when that stress is beyond the elastic limit of the material. In reply to the statement that "the ultimate strength interests us just as much as the elastic limit," I would say that I believe designers will be treading on much safer ground when they confine themselves to proportioning parts with a factor of safety based on the elastic limit rather than the ultimate strength.

4 Mr. Ellis says in the second paragraph "We note primarily that in the derivation of his formula the writer has assumed the entire area to be in tension, i. e. the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks it will seldom be encountered in shear housings." No such assumption is made, nor is it universally correct in hooks. I believe that Mr. Ellis is also mistaken in his remarks on the particular form of the equation when ρ is infinite. When ρ is infinite the case is not that of simple tension but rather as expressed by Unwin's formula.

5 Mr. Loring's explanation of the two analyses, I regret to say is incorrect. Both analyses are founded on a determination of the relation between *unit* stretch and intensity of stress, but the real difference is found in the methods of evaluating the unit stretch. The older formula gives the *unit* stretch as

$$\lambda_y = \lambda_\beta + \frac{y'}{\rho'}$$

while the newer analysis gives

$$\lambda_y = \lambda_\beta + \frac{\dfrac{y'}{\rho'} - \dfrac{y_0}{\rho_0}}{1 + \dfrac{y_0}{\rho_0}}$$

where

λ_y = unit stretch of any fiber a distance y^1 from the gravity axis.
λ_β = unit stretch at gravity axis.
ρ' = radius of curvature at gravity axis after stretching.
ρ_0 = same before stretching.
y_0 = modified y' after stretching.

The newer analysis retains terms of the same order of magnitude as λ_y and therein lies the difference. The theory of lateral contraction is rationally applied in this analysis, its application being unnecessary to the test piece, since direct measurement of stress is made.

6 Par. 2 in the paper is obviously limited to curved members. The similar form referred to in Par. 8, includes the radius of curvature.

7 In Fig. 4, $k^2 = 68.56$. In Par. 13 the equation should be $\gamma_2 = \frac{k\,e}{0.7\rho^2}$. The method uesd for determining γ_1 and γ_2, I believe will be found more convenient than those proposed by Mr. Loring.

8 Professor Houghton will agree with me that a more correct analysis for straining action will permit a more intelligent use of the factor of safety.

9 Mr. Myers is quite correct in his remarks on the value of ρ_0. The analysis, however, does apply to the case of straight beams where $\rho_0 = \infty$, for which case it reduces to the form of the Unwin formula. The formula has not as yet been sufficiently developed to determine its usefulness in establishing proportions for other than those sections at right angles to the load. The difficulty of determining the stretch on sections at an angle to the load will leave this problem unsolved for some time. It is, however, rational to suppose that the flange on oblique sections should be thickened, but to what extent has not yet been determined. In regard to the behavior of a T-section, I would state that Professor Pearson has found experimentally that it is subjected to the same laws as a solid section. This indicates that the suggestion of Mr. Myers in Fig. 4 can hardly be accepted.

10 I judge from Professor Burr's remarks that he discredits the analysis by Professor Pearson on the basis that it is founded on the common theory of flexure, that is, it is not applicable to beams of very great depth compared with the length. I believe that if Professor Burr had given more thought to the matter he would not have made this statement. In view of the experimental results obtained by myself and others in verification of the theory and the lack of any data in verification of Professor Burr's statement, I am still inclined to believe that Professor Pearson's analysis is correct.

VENTURI TESTS FOR BOILER FEED

By C. M. Allen, Published in the Journal for Mid-October

ABSTRACT OF PAPER

The object of these tests with the venturi meter was to determine how well adapted it would be for use in measuring the feed to a boiler, in view of the variety of conditions under which it might have to operate. The methods of pumping the water through the meter, the different temperatures of the water pumped, various and fluctuating pressures and velocities of flow,—any one or several of these conditions might be met with in actual service, and the results obtained indicate that such occurrence would have practically no effect on the satisfactory performance of the work of the meter.

However, there are limits to this satisfactory operation of any one meter, and the lower limit for this size seems to be reached when the velocity of the water through its throat becomes lower than about 10 ft. per sec. In case the desired amount of water is smaller than the quantity which would produce this velocity in the meter, a smaller meter would be installed. It is evident from these tests that the venturi meter is sufficiently accurate for the majority of commercial or engineering requirements.

DISCUSSION

F. N. Connet. I think that the coefficients shown by Fig. 2 are slightly less than they ought to be, because I suspect that the "venturi head" was considered equal to 13.6 times the difference of the mercury levels in the manometer, whereas this ratio is actually 12.6. Correcting for this difference would slightly increase the coefficient and would make it correspond more closely to those obtained in our own experiments.

2 The correction necessary for difference in temperatures is not as great as with mechanical meters, for the reason that the venturi meter itself automatically compensates for one-half of the difference in specific gravity. In other words, if the water be hot and the specific gravity 2 per cent less than that for which the meter was calibrated, a correction of 1 per cent is automatically made by the meter and therefore a further correction of only 1 per cent is necessary, whereas, with a mechanical meter depending upon volumes, a correction of 2 per cent would have to be made if the readings were desired in pounds. The

reason for this difference between the two types of meters is that the flow through the venturi meter is proportional to the square root of the venturi head and is not directly proportional to it.

3 The only reason for not always obtaining accurate results with the venturi meter for boiler feed is the presence of severe pulsations in velocity due to the action of the feed pump. The most accurate results can be obtained when the feed pumps are of the centrifugal type and many such pumps of the two-stage or three-stage turbine variety are now in successful use. The pulsations which are due to the action of the water plungers or to defective valve action in a reciprocating pump, make it necessary to place a rather large air chamber directly on the pump, or on the feed line as close as possible to the pump. If placed on the feed line, it should not be connected on a tee set in the line but it should be so arranged that all of the water will pass through it. Furthermore, the cross section of such an air chamber should be large and the arrangement should be such that the surface of the water will rise and fall with each stroke of the pump, There should be a gage glass on the side of the air chamber so as to insure the presence of a sufficient vacuum of air. These precautions will render the velocity of the water sufficiently uniform to obtain accurate results with the venturi meter.

4 I notice also that the results obtained were not very satisfactory when pulsations were present and when the throat velocity was less than 10 ft. per sec. There were three reasons for this which seldom if ever exist in actual venturi meter installations:

a The instrument used in the test was a mercury U-tube or manometer, containing but little more than a pound of mercury. The inertia of the mercury was therefore small and the mercury levels were unsteady. In an actual installation a registering instrument is generally used which contains almost 100 lb. of mercury, the mere inertia of which has a decided "damping" effect.

b The graduations on a manometer scale are quite close together at low throat velocities. At 10 ft. per sec. throat velocity, the difference of mercury levels is only 1½ in. In the registering instruments the movements are increased by a lever so that accurate readings are facilitated.

c During the tests described in the paper, the globe valves in the two pressure pipes were partially closed to minimize the mercury level fluctuations, and in all probability the valve discs were slightly loose on the valve stems. This therefore allowed the discs to behave like check valves

and permitted a freer flow in one direction than in the other; consequently incorrect mercury levels would result.

5 There are at least three better ways to throttle one or both of the pressure pipes than by using globe valves. The first and perhaps the best way is to use a capillary tube, say $\frac{1}{8}$ in. inside diameter by two or three feet long. The second way is to use a needle valve which is similar to a globe valve, but without a loose valve disc and with a long tapered point directly on the valve stem. The third way is to use a cock instead of a valve. Any of these methods of throttling combined with an ample air chamber permits accurate venturi meter readings at throat velocities as low as 2.8 ft. per sec. This extends the range of the meter from its maximum capacity down to one-thirteenth of the maximum.

6 Although a manometer, because of its portability and simplicity, is particularly well adapted to the making of short boiler tests, it nevertheless is not automatic and it shows the rate of flow only at the moment of observation, and if this rate fluctuates considerably from minute to minute, it becomes necessary to take very frequent readings. For this reason an instrument has been perfected which has two dials, one for indicating the rate of flow and the other for continuously recording this fate upon a circular chart paper. A special planimeter enables the charts to be measured so as to obtain the total quantity of water. This planimeter multiplies the factor of velocity by the factor of time and the product, of course, represents quantity. This type of recording instrument is largely used for meters 4 in. and smaller in diameter but for larger size meters the users generally prefer a three-dial instrument of the integrating type in order that the total quantity of water may be read directly upon a revolution counter without the aid of the planimeter.

Clemens Herschel. Professor Allen's paper shows, by tests properly and skillfully made, that the meter is reliable for hot water and boiler-feed service, and is new and unique as reproducing in tests of the meter the curious conditions to which a boiler-feed water meter is subjected. But for this feature the tests would have been only a repetition of other tests already made. Not that such repetitions are not desirable, especially when made as accurately and with the scope and purpose of those given in Professor Allen's paper. Further series of rests on venturi meters of all sizes, are in fact still called for in the interests of exactitude. But they can only in a general way confirm, not discover.

2 The point to be considered is, that several thousand venturi water meters are now in use, the world over. They are the embodiment of the action of one of the laws of nature, and are but little dependent on a correction by coefficients. They have been tested in various sizes, from $\frac{1}{4}$-in. to 10-ft. main pipe diameter, and operate exactly alike in all these sizes. They are also used to meter gases, brine and chemicals, and, as we see from the paper, to meter hot water. It is indeed a curious circumstance, that while the inventor and the manufacturers of the venturi water meter never expected to see many of these meters of less than 12 in. diameter used in practice yet the demand for hot-water boiler-feed meters has exceeded in value that of all the other sizes, for certain periods.

Sanford A. Moss. I understand from Par. 7 that the discharge of the venturi meter was figured on the basis of cold water with standard density in all cases, and that the theoretical effect of change of density was not taken into account in the formula. This would mean that Professor Allen's curve takes account of the effect of density changes, as well as all other changes. The actual formula used, and a sample of the calculations, might be a desirable addition to the paper.

2 Assuming that the above interpretation is correct, Professor Allen's curve shows that the actual flow in pounds per hour, with a given pressure, increases as the density decreases, due to rise of temperature. Is this not surprising? Theoretically, flow should decrease with the square root of the density. Of course change in the orifice friction coefficient, due to change of density, temperature, etc., might occur to such a great extent as to overbalance effect of density change. The actual orifice friction coefficient would then have a greater upward slope than in the chart so as to be over .98 per cent at 200 deg. Orifice friction coefficients for all density conditions and all fluids are usually the same for velocities occurring in practice, which are always above the "critical velocity" where fluid adjacent to a wall is stationary and where viscosity is a factor. Thus the orifice coefficient for air is the same as for water, even though the density is decreased about 800 times.

F. N. Connet. If I understand Dr. Moss correctly, he states that the quantity decreases as the density increases. With the venturi meter this depends upon the character of the graduations. If the units are cubic feet the readings *decrease* in proportion to the square

root of the increase of density, but if the units are pounds the readings *increase* in proportion to the square root of the increase of density. One is exactly the reverse of the other.

Geo. A. Orrok. I note that Professor Allen has obtained results for the coefficient of the venturi meter similar to those given by Clemens Herschel in his paper presented before the American Society of Civil Engineers, December 21, 1887, the lower values of the coefficient appearing at a velocity of about ten feet per second.

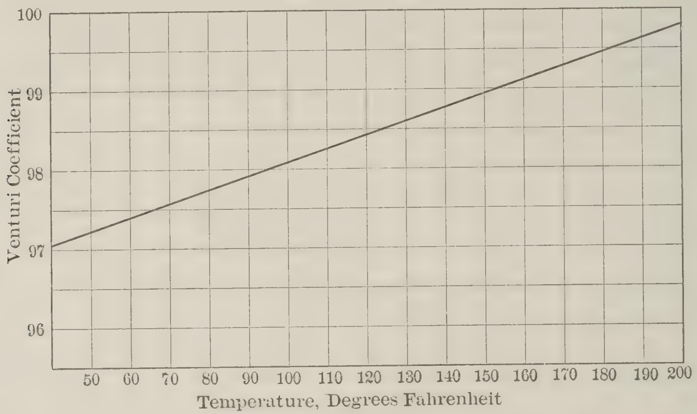
2 The New York Edison Company for some years has been using venturi meters for the measurement of water. We find them accurate and very convenient. For the last three years we have been using them in the testing of our boilers, having conducted a series of check experiments to determine the variations with temperature. Our condition is considerably better than Professor Allen's, since we use centrifugal feed pumps and consequently have a steady reading on the manometer.

3 In cases where we have both weighed and measured the feed water our results were remarkably close. On a 7-hr. test, where about 170,000 lb. of water was fed to the boiler, the meter exceeded the weighing by 631 lb., or approximately 0.37 of one per cent. In another test, in which nearly 200,000 lb. was fed, the difference was about 0.47 of one per cent. I believe the meter readings are more nearly correct than the weighing, as there was considerable opportunity for evaporation from the tanks in which the weighing was done.

The Author. Mr. Connet is correct in his statement concerning the coefficients of the venturi meter, relative to temperatures as shown in Fig. 2. There should be a correction. Each coefficient should be multiplied by the factor 103.8. This raises the coefficient to much nearer unity. The curve herewith shows the relation between the values of the coefficient for the varying temperatures.

2 I agree with Mr. Connet in regard to the throttling of the water in the pipes leading to the manometer. I believe the needle valve, or a fairly long pipe of small diameter, would be a decided improvement over the globe valves which were used in these experiments. We had not discovered that the movement of the end of the globe valves affected the reading, but Mr. Connet has had a good deal more experience along these particular lines, and I am perfectly willing to believe that this is true and that these fluctuations could be materially cut down and yet give the true mechanical average. This is

what we are looking for, and it is a good deal better than using maximum and minimum readings and then obtaining the arithmetical average. The mechanical average obtained by means of throttling is certainly more accurate because we do not know how long the maximum deflection continues, relative to the minimum.



Curve Showing Variation of Venturi Coefficient with Rise in Temperature

3 For the benefit of Mr. Moss, I would state that the density at different temperatures was considered. The following is a sample test giving an idea as to how computations were made:

If W = actual weight of water from weighing tank, then

$$W = 60\,w\,a\,C\,t\sqrt{2gh}$$

w = weight per cu. ft. at the temperature
a = area venturi throat
C = venturi coefficient
t = time in minutes
h = venturi head

$$C = \frac{W}{60\ w\ a\ t\sqrt{2\,gh}}$$

$$C = \frac{W}{1.48\ w\ t\sqrt{h}}$$

DATA OF TEST

Time 3:40 – 3:51; duration 11 minutes.	lbs.
Weight of tanks at beginning	1158
Weight of tanks at end	5369
	4211
Deduct for tank calibration.	20
	4191
Add for evaporation	2
Total water	4193
Mean mercury deflection	17.24 in.
$h = 1.05 \times 17.24$	$= 18.1$ ft.
$\sqrt{h}$	$= 4.25$

w for temperature of 137 = 61.43

Weight = $1.48 \times 11 \times 61.43 \times 4.25 = 4250$

$C = \dfrac{4193}{4250} = 0.986$ coefficient of venturi meter.

COOLING TOWERS

By J. R. Bibbins, Published in the Journal for Mid-November

ABSTRACT OF PAPER

Developments of recent years of both steam and power plants have demonstrated the usefulness of the cooling tower. This piece of auxiliary apparatus has always been more or less neglected.

It is true that in some plants the maximum effectiveness of the cooling tower and that of the condensing plant are in a sense diametrically opposed—one profits by the shortcomings of the other. The tower works best when the vacuum is lowest. On the other hand this tends to a general operative equilibrium and often saves the day when two interdependent types of equipment would succumb. Fortunately, improvements in condensers is being actively pushed, the trend being to secure higher hot-well temperatures. This immediately enhances the effectiveness of the cooling tower. Similarly, in gas-power plants, the possibility of cooling jacket water by means of this apparatus is favored by the high temperatures of discharge which prevail in engines of good construction. It is not an impossible state of affairs for the jacket water in a gas-power plant to cost more than the fuel, if not cooled and used over again, so that from all standpoints the cooling tower is worthy of careful study.

It is the object of this paper to bring into concrete form for discussion the most prevalent ideas in cooling tower construction, and a simple, inexpensive type employing lath mats is suggested together with suggestions for a combination of natural draft and forced draft types.

The performance data included in the paper are merely to give some idea of the general characteristics of the latter type of cooling tower under various conditions of operation, rather than to represent the results of a highly scientific test.

DISCUSSION

Geo. J. Foran. Evidently Mr. Bibbins has intentionally restricted his discussion to the subject of the paper, the cooling tower. He has, however, presented certain tables which, without discussion, are liable to be misleading with reference to the condensers and general cooling-tower condensing situation.

2 The paper discusses the tower quite fully, but classifies the condenser as good, bad or worse without discussion. This is made possible by assuming that the various condenser results obtained are simply a question of condenser design. This permits the inference to be drawn that the various results can be obtained at the same, or practically the same, cost, which is incorrect. Some of the results stated are possible of attainment, but would not show profitable investment.

3 It is impossible to differentiate the tower and condenser quite so completely as in the paper. Each is strongly influenced by the possible range in operation of the other, and I would like to show just how the relative sizes and consequent costs of the plants will be modified by the results desired.

4 Observers agree that the heat transferred through condensing surface varies directly with the mean temperature difference between the two sides of the tubes. Whether this mean should be arithmetical or geometrical is immaterial for the present discussion, and for simplicity I have selected the arithmetical mean.

5 It is unnecessary to assume condensers of varying grades of design and efficiency; in fact, it hopelessly complicates the question, and for my discussion I have assumed a condenser of uniform design and maximum efficiency with a varying amount of surface, which will permit us to obtain the various results tabulated by Mr. Bibbins.

6 The fairly universal practice for high-vacuum work for the past few years has been that for a 15-deg. rise in temperature of the incoming circulating water, during its passage through the condenser, it will be brought to within 15 deg. of the temperature corresponding to the vacuum. The proposition is frequently made to add only 10 deg. to the water and bring it to within 10 deg. of the vacuum. This is perfectly feasible, but we must see what this involves.

7 It means, first, that if we must carry away the heat from the steam by increasing the temperature of the circulating water 10 deg. instead of 15 deg., we must have 50 per cent more water with consequently larger and more expensive circulating plant and piping. With a 15-deg. rise to within 15 deg. of the vacuum temperature, the mean temperature difference between the steam and water side of the tubes will be 22½ deg. With a 10-deg. rise to within 10 deg. of the vacuum temperature, the difference will be only 15 deg. or, in the latter case 50 per cent more surface will be required.

8 Following the 28-in. vacuum line in Fig. 7, it will be noted that Mr. Bibbins has added practically 15 deg. to the condensing water and has given three curves—one for a good condenser with a temperature difference of 10 deg.; a very efficient condenser, 5 deg.; a perfect condeser, 0 deg.

9 Let us consider only the perfect or maximum-effect condenser with varying surface to produce the results named. For the zero-degree curve the mean difference between the steam and water side of the tubes will be 7½ deg; for the 5-deg. curve this becomes 12½ deg. and for the 10-deg. curve, 17½ deg. Or, if we should take the case

where we add but 10 deg. to the water, these three mean differences would become 5 deg., 10 deg., and 15 deg. respectively, so that the condenser for the zero-degree curve would have twice the surface required by the condenser on the 5-deg. curve and three times the surface required for the 10-deg. curve.

10 While there are several plants which report a circulating delivery temperature at approximately the temperature of the vacuum, it is evident that no plant should depend upon such a performance to obtain the economical results upon which the plant investment is based, as this would require absolutely perfect test conditions in every day operation; it would give no leeway at all and would result in too wide a variation in performance for a slight falling off in operating efficiency. Even a slight air leak would result in lowering the temperature in the vacuum space 5 deg., with a consequent loss in heat head and reduction in heat transference, owing to the presence of the air itself. These matters must be considered in addition to the question of cost.

11 Again, following the 28-in. vacuum line in Fig. 7 until it intercepts the 10-deg. curve, it will be found that it calls for water at 75 deg., the 5-deg. curve calls for 80 deg. and the zero curve for 85 deg. All these conditions assume that these results depend only upon the condenser, and if I understand the table correctly, call for the same quantity of steam and water, the temperature of the circulating water, it will be noted, being raised 15 deg. in each case. The author also assumes that the water is cooled to the temperature of the outside air.

12 Although I am sure that the author does not intend to convey the apparent meaning, the further statement is made that this calls for a fixed cooling tower performance; in other words, as I understand it, that the size of tower and the performance will be the same, to cool a given quantity of water through the same range in temperature, irrespective of the temperature of the air.

13 Let us follow this a little further, and in line with the general assumptions, assume for this purpose that the hot air leaves the tower at the temperature of the hot water and 100 per cent saturation. By reference to psychrometric tables it will be seen that each cubic foot of air at 70 deg. temperature and 70 per cent humidity, when increased to 85 deg. and 100 per cent, will take on 7.15 gr. of moisture, whereas a cubic foot increased from 47 deg. and 70 per cent to 62 deg. and 100 per cent, will take on only 3.575 gr. of moisture; that is, although the temperature is increased 15 deg. just the same, the air carries away

but one-half the moisture at the lower temperature, showing that twice the air capacity of tower efficiency will be required at the lower temperature. This is better understood when we consider that within the usual air temperature ranges, the moisture-carrying capacity of the air is doubled for each 22-deg. rise in temperature. To be brief and to avoid confusion, I have used the ordinary nomenclature, which is scientifically incorrect. We all understand that it is the space and not the air which is saturated, but this splitting of hairs would not affect the point under discussion.

14 I have purposely neglected the several minor considerations as they affect the question to a very small extent. For example, the volume of the air entering the tower at 70 deg. and 70 per cent humidity, and leaving at 85 deg. and 100 per cent humidity, is increased nearly 5½ per cent, due partly to the increased temperature and partly to the reduced pressure of the air itself, owing to the increased saturation and vapor present. It is well known that the cooling tower performs its work *principally* by the withdrawal of heat from the main body of water which provides the latent heat for the evaporation of a small portion of the water carried away in the form of vapor as increased humidity of the cooling air.

15 Temporarily omitting the *perfect* plant, let us consider an *average* operating plant in a location having air at 70 deg. and 70 per cent humidity. The usual cooling-tower turbine plant would carry a vacuum of 27 in. with water cooled from 100 deg. to 85 deg. If it is desired to cool this water from 90 deg. to 75 deg., this would permit of carrying a vacuum of 27¾in . with the same amount of surface and water, but would require an increase in the quantity of air and of tower capacity of approximately 50 per cent. If it is desired to cool the water through only 10 deg. that is, from 85 deg. to 75 deg. and to bring the water within 10 deg. of the vacuum (28¼ in.) this would call for 50 per cent more water, 50 per cent more surface and over 100 per cent more air and cooling tower capacity than for the usual 27-in. vacuum plant.

16 There are hardly two plants which have quite the same determining factors. The determination as to the advisable vacuum and plant must be decided in each case, but there are few plants where the conditions would warrant the installation of a plant to produce the maximum vacuum under the most severe conditions.

17 With reference to the type of tower with fans in the stack, as shown in Fig. 2, the Worthington Company installed their first tower of this type with rope fan drive, in 1900, and recent reports indicate as

good results as when the tower was installed. As a general proposition, however, there are several questions to be considered in comparing this type. There is a saving in the number of fans over the arrangement with the fans below the tower filling, but the fan operates in the hot, highly saturated air, is more or less inaccessible and out of sight, and therefore will not receive the best of attention. It requires good installation and is more difficult to maintain in good condition owing to the fact that it is an exhauster. Any of us would prefer to install a pressure fan rather than an exhauster; the capacity of the fan in the stack must be somewhat larger for the reason that as neither the circulation or the surface efficiency is improved, the total volume of free air required is the same, this being handled at a less pressure and higher temperature and humidity.

18 Comparing the fan and natural-draft towers, there are few, if any, locations where high results are desired, where the natural-draft tower could be selected. A little calculating will convince any engineer that the draft is principally due to the wind velocity over the tower. Study of the meteorological tables will show that in most power centres, except in very few locations, the wind velocity is much greater in winter than in summer—just the opposite of our requirements. This is clearly demonstrated in the operation of any fan tower from the fan speeds permissible at different seasons. It must be remembered that with a tower of the same height the wind assistance is the same for either type of tower. There are many locations where a so-called combined tower can be used if the additional expense is warranted, but strictly speaking, the operation cannot be combined. It must be used either as a natural-draft tower or as a fan tower, but if the fan is operated at all, all the air must pass through it, whether the fan is located above or below the filling.

19 I do not see how there can be any induction in the tower shown in Fig. 14. The object of the tower is to get sufficient pressure below the filling to force through the requisite amount of air, but this pressure must be uniform in the entire space below the filling in order to obtain complete surface efficiency, and under such conditions air would leave rather than enter the tower through any additional openings to the outside air.

20 The Worthington Company make a so-called combined tower which permits of two water levels in the cold well. At the lower level the air enters through the fan at rest and below the lower plates of the tower shell above the water. At the higher level the lower plates are sealed and all the air enters through the fans, which can be operated

at the speed necessary to supply the additional pressure required by the low wind draft. This is also accomplished by the use of additional draft doors.

Prof. William D. Ennis. Will Mr. Bibbins explain in more detail the derivation of the curves in Fig. 7? The tower must provide cooling sufficient to absorb the heat liberated with the exhaust steam, viz., 939 B.t.u. per pound. The amount of cooling in each case would then be 939 divided by the weight of circulating water per pound of steam. On this basis, the maximum temperatures of entrant air agree closely with the curves at 27-in. and 28-in. vacuum, but are about 1 deg. higher than the curves indicate at 29 in., and 2 deg. or 3 deg. higher at 26 in. The curves should apparently be more nearly straight.

2 The paper gives unusually complete and valuable data on many phases of cooling tower operation, but it is to be regretted that the matter of loss of water has not been dealt with in more detail. This is perhaps the most vital question. Manufacturers are sometimes asked to guarantee a limit of loss, but it would be just as logical to ask for a guarantee as to the value of π. A rough estimate often offered is that the loss will not exceed the amount of boiler feed water.

3 Mr. Bibbins gives data from three plants: that at Duquesne, in which the makeup water was from 10 to 20 per cent; the Potosina plant, in which the loss of vapor by windage was occasionally as much as 10 per cent of the volume (of water?) passing through the tower; and the Detroit natural-draft plant, in which the vaporization loss was 2 per cent of the water passing through; practically equal to the weight of boiler feed. The average cooling per hour was $(293{,}530 + 5910.6) \times 16.23 = 4{,}860{,}018$ B.t.u. Each pound of water vaporized, if we neglect the cooling effect of the air, must then have absorbed $\frac{4{,}860{,}018}{5970} = 816$ B.t.u. This is the nearest to a reasonable result I have ever seen in a cooling-tower test.

4 Usually, and this apparently applies to the two other cases cited by Mr. Bibbins, the loss of water is far greater than theory indicates as necessary. The cooling of the water is accomplished by (*a*) the absorption of heat by the air and (*b*) the evaporation of a portion of the water. When the minimum temperature of the air equals or exceeds the maximum temperature of the water, the first effect becomes zero. When the air is initially saturated, the second effect becomes zero, except as the air is heated during its passage. Under

the limiting condition at which there is no direct transfer of heat to the air, the necessary volume of air is increased, and the loss of water does all of the cooling; but the proportion lost need not exceed, in theory, the quotient of the range of cooling by the heat of vaporization, and the use of screens unables us even to reclaim some of the otherwise lost vapor. Why is it that almost invariably the make-up water greatly exceeds the amount thus computed as necessary? It is inferred from Par. 34 that Mr. Bibbins has considered this question of cooling by evaporation, in which case some exposition would be desirable.

Henry E. Longwell. Very early in 1884 under the direction of John C. Dean, of Dean Brothers Steam Pump Works, I made drawings for a cooler that was built for the Kane Milling Co., Kane, Ill. I am told that it was the first one erected in the United States, and it is, at any rate, a well-authenticated case of a very early installation. The plant was operated for only two years, being then destroyed by fire, but so far as I can remember the installation performed in a very creditable manner, especially considering the primitive state of the art at that time.

2 There are probably many engineers who will take issue with the author if he means that the cooling tower field is yet comparatively unexplored. For ten years or more the cooling tower has been on a strictly scientific basis. Its design and construction constitute a branch of engineering that is just as distinct and as well developed as any of those which deal with other specialties such as gas engines, steam turbines and the like. When we consider that one builder alone has constructed about 2000 cooling towers which in the aggregate are capable of cooling condensing water for about 3,000,000 horsepower, we must admit that this device has progressed a long way beyond the rudimentary stage.

3 It is not excessive cost or lack of knowledge that has restricted the use of cooling towers in the United States. It is because nature has been so good to us that the conditions in which cooling towers are desirable or necessary are comparatively rarer than in the less favored and more congested European countries, where these devices have reached the highest state of development.

4 I regret that the author has not presented in exactly the same form the two tests of the cooling tower described. In Table 4 is given a complete log of the principal observations made at approximately hourly intervals; in Table 5 we have only the average of all the

observations made over a period of 24 hours. The two tests were made under such widely different conditions that they afford no proof as to whether the performance of the tower was any better or even as good with its full complement of cooling surface, as it was with only three-fifths of it. During the test with only three-fifths of the cooling surface installed, the average load was nearly 80 per cent greater, and the average quantity of water circulated per hour was nearly 35 per cent greater than on the test with all of the surface installed.

5 Referring to Fig. 11, the indications are that the added cooling surface served no useful purpose. Indeed if the diagram means anything at all, it means that for the same temperature head the product of the heat dissipated per square foot of surface per hour multiplied by the proportion of the cooling surface installed, is practically a constant; also, that for equal temperature heads, the number of degrees cooling is practically the same.

6 In Fig. 12, in which temperature head is plotted against degrees of cooling, the lines corresponding to 3/5 surface and 5/5 surface, coincide so nearly that one could hardly say that they depart from each other by more than the limit of the normal error of observation.

7 Fig. 13 at first sight seems to indicate that at hot-well temperatures below 120 deg. the cooling was considerably greater with 5/5 than with only 3/5. But we know that on the test with only 3/5 of the surface, the amount of water circulated was very much greater than with 5/5 surface. Comparisons of this sort are misleading unless the quantity of water circulated per hour and the temperature of the incoming air are the same in both cases.

8 The inconsistency of the curves in Fig. 13 will become apparent if we extend the straight line curve for 3/5 surface until it cuts the line of zero cooling. This will indicate that at a hot-well temperature of a little above 85 deg. the water would not be cooled at all, although we know from Table 4 that the temperature of the incoming air was at no time higher than 35 deg. The inference would be that water entering the tower at a temperature below 85 degrees would be warmed by coming in contact with air at or near the temperature at which water freezes.

9 The indications are that the tower is too small for the work, and that its performance is limited, not by the amount of cooling surface, but by the weight of air that can pass through it in a given time. After all, it is the air that carries off the heat, and the quantity of air passing through the tower is just as important a factor as is the area of the cooling surface.

10 The tower described occupies 200 square feet of floor space, and is rated at 900 h.p. Assuming 15 lb. of steam per h.p. hour, the tower would have to cool sufficient water to condense 13.500 lb. of steam hourly. A natural draft tower designed by one of the most experienced builders of this class of apparatus, would for this same duty occupy a space about 29 by 24 ft., or nearly 3½ times as great as that occupied by the towers described. It would also be from 7 to 10 ft. higher, which would give a more powerful draft.

11 Referring again to Fig. 12, it will be seen that the temperature of the water leaving the natural-draft tower is from 40 to 70 deg. above that of the incoming air. On this same diagram are curves which purport to show the performance of the forced-draft tower briefly referred to in Table 3. It would appear from these curves that the forced-draft tower under favorable weather conditions cools the water to within 3 or 4 deg. of the atmospheric temperature. Under unfavorable weather conditions it appears to cool the water to within 15 to 35 deg. of the temperature of the atmosphere.

12 The cost of the forced-draft tower is given as \$2.60 per h.p. as against \$1.50 for the natural-draft tower. However, if the comparative results as shown in the diagram Fig. 12 are dependable, it would appear that the forced-draft tower was well worth the additional cost, and a little bit more.

13 In Fig. 7, the author purports to show the maximum temperature of inlet air permissible for various vacuums. This diagram really shows the maximum temperature of cooling water to produce a given vacuum on the assumption that we limit the number of pounds of cooling water per pound of steam condensed, to the arbitrary figures set down in the lower right-hand corner of the diagram. The temperature of the atmosphere is not necessarily the limiting temperature to which the water may be cooled. It is well known that with low humidities, cooling towers may reduce the temperature of the water to several degrees below that of the atmosphere. And there is no law of nature that stipulates that we may circulate no more or less than 100 lb. of condensing water per pound of steam to produce a 28 in. vacuum, or 60, 40 and 30 lb. per pound of steam to produce respectively vacuums of 27, 26, and 25 in.

14 Fig. 15 is doubtless interesting, but as no reference to its usefulness appears in the paper it is difficult to see wherein it is pertinent.

15 I would point out that the diagram **Fig.** 1 shows that on two days in June 1906 the *average* temperature exceeded 90 deg. According to Table 1 on the following page there was not a single day during

that month on which the *maximum* temperature reached 90 deg., to say nothing of the average. If there were 10 days in the month of June 1906 on which the temperature exceeded 75 deg., it is difficult to see why there must not have been at least as many days on which it exceeded 70 deg. The quantities set down in the columns headed "Average for Month" require some explanation to make them intelligible.

16 The theory of cooling towers is simple, and any one who has a reasonable acquaintance with that branch of natural science which deals with heat, may easily know it a little or even very well. As far as the theory itself is concerned it would be hard to improve on the clear, concise and generally masterly presentation of the subject by F. J. Weiss, inventor of the well-known Weiss condenser, which appeared in a book entitled "Kondensation," published in Germany about ten years ago. But as in all branches of engineering, the coefficients by which theory is reduced to practice, are the property of the few, who by special application and practical experience have come to know the subject profoundly.

Barton H. Coffey.[1] The advent of the turbine with the high cost of fuel in steam plants and the increasing cost of water for cooling purposes in urban installations of refrigerating apparatus, are making the cooling tower a necessary means of economy.

2 As the author remarks, the literature upon the subject is scanty; in fact with the exception of C. O. Schmitt's paper before the South African Association of Engineers in 1907, there is scarcely anything extant that I know of, worthy of the name.

3 I do not wholly agree with Mr. Bibbins' presentation of the meteorological conditions to be met by cooling towers, as given in Fig. 1 and Table 1. The comparison of average humidity and temperature, as given by the weather bureau, is a little misleading, as the humidity observations are made at 8 a. m. and 8 p.m. only. In lieu of hourly humidity measurements, I think it better to take the average aqueous pressure at 8 a.m. and 8 p.m., as it is known that this quantity changes slowly, and from this the hourly humidities can be calculated. It will then be found, of course, that as the temperatures advance toward midday, the humidity falls, thus tending to maintain average thermal conditions with respect to cooling towers and explaining the approximately uniform results actually obtained. The mean aqueous pressure for July, covering a number of years, work out about as follows:

[1]With Edwin Burhorn, 71 Wall Street, New York.

Table 1 Mean Aqueous Pressure

	Actual Aqueous Pressure
Boston, Mass	0.542 in. mercury
Philadelphia, Pa	0.614 " "
Salt Lake City, Utah	0.296 " "
St. Louis, Mo	0.648 " "

At St. Louis, therefore, where the mean maximum temperature for July is 88 deg., the relative humidity would be 49 per cent against a mean humidity of 66.1 per cent, as given by the tables, which is distinctly a more favorable condition for cooling towers.

4 While on meteorology, I would like to call attention to the statement in Par. 15*b*, that the tray or atmospheric type of tower cools only by means of "transverse air currents from the side", the obvious deduction being, that without wind this type of tower fails. As a fact, in a dead calm the efficiency of all forms of tower falls off, but this condition is of small practical account, as in the interior region the percentage of calm rarely exceeds 2 per cent of the month and on the seaboard is practically unknown. However, in a dead calm the towers still continue to work, due to an ascending column of warm air and aqueous vapor over the tower and a corresponding horizontal inflow of cool dry air. This condition must exist, otherwise the entire space surrounding any tower would become filled with warm saturated air and all cooling would cease. In a forced-draft tower for example, the fan would be simply circulating air having no capacity for absorbing heat. Apropos of this, I have records from an atmospheric tower on refrigerating work for the entire month of September 1907, taken with recording thermometers, in which the cooling water from the tower was maintained at an average of 75 deg., never exceeding 80 deg., with a cooling range of about 10 deg.

5 In Par. 13*b*, among the elements of design, Mr. Bibbins advises "Avoid free falling water. It should be distrubuted so as to descend clinging to some form of wetted surface." I would like to know the basis for this statement, as probably by far the largest number of towers in use throughout the world employ the principle of finely divided falling water, as, for instance, the various forms of atmospheric and chimney towers in Europe, South Africa and this country.

6 As 75 to 85 per cent of the cooling is due to evaporation, which can take place only at the surface in contact with the air, the form of cooling surface is of great importance. In a cooling tower with free-falling water, the cooling surfaces consist of the hurdles or decks and the exposed surface of the falling water. Experiments show the

weight of a drop of water to be about three-fourths of a grain, the diameter of the corresponding sphere being 0.178 in. A gallon of water properly distributed will therefore expose about 54 sq. ft. of surface. If we know the flow per second and the time of fall in seconds, properly corrected for atmospheric retardation, we can calculate the exposed surface in the water, which, added to the fixed wetted surface, gives the total cooling surface in the tower. The efficiency of the surface in the falling water is greater than the fixed surface, due to the greater velocity of the air relative to the water surface, due to the motion of the drops.

7 The question of type of surface, in my opinion, is one of expediency to be determined by the conditions of operation. Fixed surface is undoubtedly more compact and when skillfully designed opposes less resistance to air currents. On the other hand, it involves weight, greater difficulties in distribution, and where oil is present in cooling water, it becomes coated, the capilarity is destroyed and the water film is reduced to streams, thus greatly lessening the water surface exposed.

8 If the atmospheric form of tower is to be employed, it is hard to conceive of any form of surface, save drops, that would be exposed to the wind from any direction; and where space is available for sufficient surface, the temperature reduction called for can always be attained.

9 In a test by the speaker of an atmospheric tower circulating 440 gal. per min., with air at 93 deg. and humidity 34 per cent, the water was cooled from 80 deg. to 74 deg., or within 3 deg. of the wet bulb, which is the limit of atmospheric cooling.

10 With reference to Mr. Bibbins' remarks on the effect of temperature range on the size of the tower, I beg to submit a few figures on the volume of air required at 80 deg. and 80 per cent humidity to absorb 1,000 B.t.u. when the air can be heated to the following final temperatures and saturated:

Table 2

Class of Work	Final Temp. Air	Cubic Feet
Refrigeration	88 deg	985
Steam Condensing 27 in. vac.	100 "	429
Steam Condensing 26 in. vac.	110 "	267

This shows the enormously increased quantities of air required as the lower ranges of cooling are approached, and also shows the particular advantage of the atmospheric tower for refrigeration work, in saving the power necessary to handle this large volume of air. As an example:

11 With air at 80 deg. and 80 per cent humidity, to cool 600 gal. of water per min. to 80 deg., would require about 70,000 cu. ft. of air per min., requiring about 17 brake horsepower in a fan tower. An atmospheric tower of like capacity, having 960 sq. ft. wind exposure, would receive 248,000 cu. ft. of air per min. at a velocity of 4 miles per hour. In steam condensing, with a limited space the forced-draft tower is, of course, the only available type.

Carl George de Laval. The author states that the present high prices constitute the greatest obstacle to the use of cooling towers, and, further appears to give the impression that the cooling tower is a makeshift and not a permanent apparatus.

2 There are three classes of towers, forced-draft, natural-draft and a combination of both, the last-named being used either way depending on the season of the year. The selection of the type should depend on climatic conditions, cost, etc., a dry climate being best suited for a cooling tower.

3 The author states that the costs range from $4.80 to $6.93 per kw., which appear to be slightly higher than market prices, the reason perhaps being that the author had imposed severe conditions when asking for bids on cooling towers, thereby increasing the costs.

4 Let us assume a plant of 1,000-kw., consuming 19,000 lb. of steam per hr., basing the condenser performance upon the ordinary 10-deg. difference in a counter-current jet condenser, and upon a 27 in. vacuum, with air at 70 deg., and 70 per cent relative humidity. A cooling tower with interlocking pipe filling can be built approximately 19 ft. by 35 ft., fitted complete with fans, for about $5 per kw., and a wood-filled tower about 21 ft. by 35 ft, for about $4.50 per kw.

5 The author is correct in stating that installations are not being sufficiently studied, and this, no doubt, is the principal cause for the failure of cooling towers and has prevented their more general adoption. It is not sufficient merely to obtain information as to maximum load, steam consumption, maximum temperature and humidity, but it is necessary to know whether these maximum load conditions must be met at the conditions of maximum temperatures and humidity, and if so, for how long a time.

6 Let us assume that bids are asked for a cooling tower for 8,000 kw., the conditions named being an air temperature of 75 deg. and 75 per cent humidity, 27 in. vacuum, no time being stated when this load of 8000 kw. is likely to occur, and what its duration is. The real facts may be that this load comes in winter only, and that in

summer probably not over 5000 kw. would be required during the evenings, while the summer mid-day load might not be over 2000 or 3000 kw. Under such conditions a tower calculated for a 5000-kw. summer load would be ample for an 8000-kw. winter load, and if the installation was made on the basis of 8000 kw. the year round, the cooling tower would be too large and expensive, and the cost per kilowatt of maximum load would be too great.

7 The maximum mid-day temperature and humidities likewise should not be the basis of consideration with maximum loads, as the electric lighting plant maximum during summer should instead be based upon 8 p.m. temperature and humidities. One sometimes sees the requirement to handle maximum loads at an atmospheric condition of 90 deg. and 80 per cent relative humidity—a condition that may never be reached in the particular locality where the tower is to be installed.

8 Most of the towers described by the author appear to be home-made or makeshift towers, for instance, the tower shown in Fig. 6, and, installed at Butte, Mont., having a cross-board filling and a wooden stack for natural draft. The design is such that it will lose considerable of its efficiency as it continues in service, and the boards, as well as the upper stack, will warp, admitting cold air above the filling and tending to kill the draft upon which such a tower depends for its efficiency. The warping of boards will also cause leakage through the sides of the tower, the leakage being carried by any strong breeze, and thrown against surrounding buildings and territory, where during winter it may freeze into a heavy mass.

9 Referring to preceding discussion on the design of towers for maximum atmospheric conditions, one will note in the temperature ranges in Table 2, for the Butte tower, that the atmosphere was over 80 deg. during less than 3 per cent of the total time of the year, so that such conditions can hardly be used as a basis for calculation. Atmospheric conditions at Pittsburg during the four months from May 15 to September 15 average approximately 70 deg, and 70 per cent, which appears to be about a standard basis for cooling towers.

10 Par. 9 refers to the use of cooling towers for handling jacket water of gas engines, the temperatures being about the same as those encountered in ice plants, and higher than in the case of steam condensation. Several installations show this temperature to be from 156 deg. to 111 deg., and 130 deg. to 80 deg.

11 Par. 10 and Par. 11 state that the difference between the theoretical steam temperature and the temperature of the circulating

water varies from 10 deg. to 50 deg. The usual jet condensers and surface condensers give about 15 deg., and cooling towers for reciprocating engines are usually based on a 24-in. vacuum, with circulating water cooled from 125 deg. to 100 deg. and an air temperature of 70 deg. and 70 per cent relative humidity. Counter-current condensers give about 10 deg. difference, the circulating water being handled under the same conditions of vacuum, with a temperature range from 130 deg, to 105 deg., instead of from 125 deg. to 100 deg., which of course gives an easier condition for the cooling tower.

12 It is a well-known fact that an efficient condenser must be installed in order to get good work from a cooling tower, it being an advantage to the tower to have the temperature of the hot water and the cold water as high as possible. For instance, taking examples of the two conditions, both 1000-kw plants consuming 19,000 lb. of steam per hr., at 24-in. vacuum, and an air temperature of 70 deg. and 70 per cent relative humidity, one plant being based on a 40-deg. difference between the exhaust steam and the outlet circulating water, which requires the water to be cooled from 100 deg. to 75 deg.; the other plant being based on a 10-deg. difference between the steam and the water, the water being cooled from 130 deg. to 105 deg. In the former case, for the same load, vacuum and air temperature, we require an interlocking pipe-filled tower, 22½ ft. in diameter by 35 ft. high, having four 96 in. fans; whereas in the latter case with only a 10-deg. difference we can do the same work with the tower 13 ft., 6 in. in diameter by 35 ft. high, having one 120 in. fan. The efficiency of the condenser therefore makes a very decided difference in the size of cooling tower.

13 Under *c* in Par. 13, the author apparently refers principally to towers with wood sides, having a wood structure within the outside boarding. It is very important that the filling must come close to the side of the tower. Particular care should be taken in erecting towers to see that pipes are first laid around the outside edge as closely to it as possible; otherwise, there will be a short circuit of cold air around the side of the tower and a loss of efficiency. This condition, while bad enough in the forced-draft tower, is much worse in towers of natural-draft type, because this air will seriously reduce the draft by mixing with and cooling above the filling the heated air upon which the draft depends.

14 As to height of working section, it is true, as the author states, that the height is important, and the distance of the elevation of the water should be kept as low as possible. A pipe-filled tower is

13 ft. 4 in. deep with a drop at the bottom of from 6 ft. to 11 ft; according to the size of the tower. With a distributor operating head of 5 ft., this gives the largest towers a maximum pumping head, plus friction in the piping, of 29 ft. 4 in. against approximately 38 ft. as required in the experimental natural-draft tower at Detroit, shown in Fig. 9. The horsepower necessary to pump the water the additional 8 ft. 8 in. in height will offset the usual fan horsepowers, making a natural draft tower of this type more expensive to operate than a fan-draft tower.

15 As to the mat of wood swelling and being thrown out of place, I would state that towers have been built with a cross-board wood filling, and four of these have been in satisfactory operation since 1904. In these towers were used 2 in. by 2 in. verticals at intervals through the filling, with the boards nailed so as to hold the filling in place and prevent distortion or formation of large open gaps through warping of the filling.

16 The cooling towers illustrated in Fig. 3, are furnished with perforated pans and have free-falling water, the sides being screened. This tower depends for its efficiency upon a cross breeze and is very inefficient in still air as the air cannot rise within the tower on account of the pans. A strong breeze will blow most of the water out through the sides of the cooling tower, in spite of the screen. The tower shown in Fig 4 occupies considerable space, and also requires additional space in the immediate vicinity because of loss of water through windage. The tower illustrated in Fig. 5 is evidently much less efficient than that in Fig. 4, because of the large amount of free-falling water. The free-falling or splashing of water is a very inefficient method of cooling. Water for proper cooling should always be brought down in contact with a surface so that it will descend slowly and thus have close and intimate contact with surrounding air.

17 In Par. 26, the author gives the total cost of the Detroit tower, erected complete and including filling, distributing pipes, foundations, etc., at \$1350. It appears that the steel work, if made of at least No. 10 gauge, would weigh approximately 20,000 lb., which at 6½ cents per pound, which is about as low a rate as mentioned, would require an expenditure of \$1300 for steel work alone. The lath filling and the work of assembling and installing this tower, would cost about \$400; the timber supports, distributing piping, etc., about \$250; concrete foundations an additional \$250, or a total cost of \$2200. Assuming a load of 1000 hp. with 19,000 lb. of steam per hour, vacuum 24 in., with a temperature difference of 10 deg., the

circulating water being cooled from 130 deg. to 105 deg., air at 70 deg. and 70 per cent humidity, a pipe-filled cooling tower of the fan type, measuring 13 ft. 6 in. by 35 ft. could be installed for about $2500.

18 The results of the test given in Table 5, with atmospheric temperatures of from 18½ deg. to 30 deg., are not complete for a natural-draft tower, as such towers fall off in efficiency very rapidly when the air temperature is raised. The results at temperatures from 70 to 80 deg., would not be so favorable.

19 In Par. 30, the condition of scale covering the wooden filling would be experienced in any tower, and is usually encountered where well water is used to make up in cooling towers for refrigerating plants. The scale forms a protecting coating in a pipe tower and prevents possible rusting of the pipe filling.

20 In Par. 32, the author refers to possible advantages of a slotted pipe as compared with spouts on a distributor arm, in regard to clogging. The spouts used by some first-class designers are 1 in. in diameter and are consequently much less liable to clog than are pipes having a ⅛ in. slot in the top.

21 In Par. 15, the author refers to the use of sprays over a pond. This seems a very simple apparatus, but it must be realized that the sprays require from 15 to 20 lb. pressure at the nozzle and so consume more power than required for circulation through a cooling tower of the fan type, and in most cases as much power as is required both for the circulating of the water and for the driving of the fan.

22 The arrangement of cascade or cooling sprays on a roof as described by the author is not new. The installation was in use by J. H. Stut of San Francisco, previous to 1892, being placed upon the roof of a factory. Galvanized troughs, 5 ft. wide were arranged in tiers on a slight incline so that the water traveled back and forth a distance of about 2,000 ft. before being returned to the condenser. An arrangement of falling from one trough to another, these troughs being spread out upon a roof, was used at the old Budweiser Brewery in Brooklyn previous to 1890. The sprays and roof troughs are alike open to the objection that if there is a strong breeze the water is carried all over the surrounding neighborhood and if there is no breeze, a heavy fog quickly collects at the point of spray and thus greatly reduces the amount of cooling.

23 Referring to various types of filling illustrated in Figs. 8*a* to 8*c*, Fig. 8*a* offers too serious an obstruction to the draft within the tower, closing more than 40 per cent of the space necessary for verti-

cal circulation of air, as against 3 per cent covered by interlocking pipe-filling or 25 per cent by wood filling. The cascades as illustrated must fall as shown in the sketch in order to operate efficiently, that is, the water must strike the pans on the next lower section of the filling, but this they will do only if the amount of water supplied is practically constant, otherwise it is liable to spill over several rows of filling, and result in quick descent and consequent loss of efficiency.

24 The filling illustrated in Fig. 8*b* is that used by Henry W. Bulkley, and depends upon a cross steam of air, as in the tower shown in Fig. 3. It is open to exactly the same objection as the latter tower.

25 The filling 8*c* will cause large quantities of free-falling water between the several courses and will result in inefficient operation. The filling 8*e*′, a wooden cross-board type, is apparently good. It requires additional expense in placing, but evidently will save something in fan horsepower. The filling 8*d* offers a bad obstruction to the draft on account of deflecting the air alternately to the right and left. The water also will evidently flow down the top side of the board, whereas the air impinges most strongly against the lower side of the board.

26 The filling 8*b* is the same as 8*b*′ and is good. The filling 8*g* is open to the objection of having no redistribution,—the water distributed at the top, however unequal it may be, must remain unequal from top to bottom. The filling 8*f* has one redistribution at the center. Otherwise it is open to the same objection as 8*g*.

27 In Par. 23, and also in the footnote, it is stated that ball bearings are difficult to keep in good condition. Ball bearings are not used in modern towers, a floating water-step bearing being used instead.

28 Referring to Par. 41 and Par. 42, a combination type tower may run with natural draft about eight months during the year. At a plant in Newark, N. J., a combination type tower with side doors operates on an 800-kw. load nine months of the year with natural draft, and requires 25 hp. during the remaining three months of the year.

29 As to Par. 46, efficient condensers are more badly needed than efficient cooling towers. Cooling towers have reached as high an efficiency as can be expected, but most plants now operating with direct jet condensers delivering into the towers, could obtain much higher vacuum or handle greater loads at the present vacuum, if condensers of the counter-current jet type, or the more efficient baffled-surface condensers were substituted in place of the condensers originally furnished.

30 As to Par. 48, the temperatures are practically the same as in ice plants. In order to get the temperature head mentioned, it is more economical to circulate the water for the ice plant first through the ammonia condenser and then through the steam condenser delivering to the cooling tower, than to have two towers handling separately the water of the ammonia condenser and that of the steam condenser.

31 The open wooden towers referred to in Par. 50 are not restricted to points of low humidity: but as already mentioned, they require much open ground, not only on account of their size, but also for wind effect, and that surrounding buildings may not be drenched with water blown from the tower.

32 As to Par. 51, the tower best adapted to natural-draft work is the one which offers the least resistance to the ascending current of air. In Par. 52, no temperatures are given to substantiate the statement of heat dissipation by lath mat construction.

33 As to Par. 53, one cannot endorse the fan booster or induction type when combination towers can be made that will give better results and that are surely preferable for overload conditions.

34 The largest number of towers in this country are of the forced-draft type, while European practice tends towards natural-draft towers. It is thus apparent that there can be no standard of type or size, because of difference in climates; each installation must be considered as a separate problem.

E. D. Dreyfus. In Par. 10, Mr. Bibbins says, "But in practice from 10 to 15 deg. difference exists, depending upon the type of condenser and the volumetric ratio of water to steam." I wish to supplement this by adding that it does not depend altogether on the volumetric ratio. Another important factor is the effectiveness of air removal. Those who have had considerable experience in condenser work find that the more effectively the air is withdrawn, the nearer the theoretical vacuum is reached. By this means it is possible to operate with a diminished volumetric ratio as the temperature rise is increased.

2 Exception is to be taken to Mr. Foran's remark that perfect condenser operation entails much greater experience, which might be implied as generally applicable. This is true only of surface condensers. In cooling tower practice, the conditions are extremely favorable to the use of the more simple jet type. The more efficiently this latter type is operated, that is, the nearer the discharge water

is brought to the temperature of the exhaust steam, the smaller is the volume of water necessary, since volume and temperature rise are component factors of the B.t.u. extraction. Therefore, with less volume of water handled, the size of the condenser may be reduced and consequently furnished at a smaller cost.

3 A remark made by the author in presenting the paper, that an inefficient condenser and an efficient cooling tower go hand in hand, bears further explanation, although the statement was modified somewhat subsequently. With an inefficient condenser, the vacuum is not likely to be very good, and therefore, with the higher temperature prevailing in the condenser, the water might pass to the tower at a higher temperature, making it easier for the latter apparatus to perform its work. On the other hand, the statements might be applied with equal, if not greater, force to efficient condensers which are able, for the same condensation, to create higher vacua, besides heating the discharge water up to the same final temperature head as the inefficient type, there being little or no terminal difference in an efficient design at its normal capacity. Moreover, considering the benefit accruing to the prime mover, a smaller volume of water may be used and worked at the same temperature as in the inefficient type of condenser, thus increasing the possibility of the tower. I would qualify the above statement to the extent that it deals with a comparison of condensers designed for the same vacuum, and evidently would not hold for a case where a very poor vacuum was admissible.

4 It might be well to state here that a near approach to the theoretical vacuum is not an impossible condition in actual operation. This implies, of course, that the character of the condenser design—the counter-current type with an efficient air pump—fulfills the requirements. In a test which I conducted last Fall on a 1000-kw. low-pressure turbine equipped with a counter-current jet condenser, the following results were obtained: At three-fourths load with 83 deg. injection water, a vacuum of 28.20 in. (30 in. barometer) was maintained, and the water left the condenser at a temperature of 96.8 deg. The temperature corresponding to the vacuum was 97.6 deg., giving practically one degree terminal difference.

5 I have observed that temperatures of the water leaving the tower were several degrees colder than the atmospheric temperature in warm weather, the difference being as much as ten degrees at times.

6 With the increasing recognition the cooling tower is receiving, it would be desirable to have the Society define a standard basis of

measuring the efficiency of the apparatus. There is a conspicuous lack of harmony of opinions as to what constitutes the governing characteristics of tower performance.

T. C. McBride In the earlier parts of the paper the author would lead us to believe that cooling towers have not received the scientific attention warranted. Reference to the literature on this subject and the work that is being done hardly confirms this fact. A considerable number of manufacturers have for some years past been supplying cooling towers designed on scientific lines, and the proposals submitted by them, particularly on fan-type towers, are intelligently framed and leave no points whatever open to guess work.

2 The paper very properly calls attention to the intimate relationship of condenser efficiency to cooling tower performance, but in doing so is extremely unfair to the condenser—in fact, in speaking of different types of air pump, the author almost leads us to believe that some are so superior to others that the vacuum they create is of a superior kind compared to that created by other air pumps.

3 Condenser engineers now agree that the efficiency of condensers, with regard to the comparison of discharge-water temperature with theoretical vacuum temperature, is as much a question of the average temperature of the vapor in the condenser as its design. The average temperature of the condenser is necessarily determined by the amount of air present therein, and is a direct function of the ratio of the air-removing capacity of the air pump and the volume of air reaching the condenser with the steam. The merit of the air pump cannot therefore be determined either from the vacuum obtained or from the relation of the discharge-water temperature to the theoretical vacuum temperature, but is wholly a question of the capacity of the air pump to handle air at the least expenditure for power, maintenance, interest on first cost and depreciation.

4 It is true that the question of condenser efficiency and air-pump efficiency is somewhat involved with that feature of condenser design having to do with the reduction of air-pump suction temperature, but as all condenser designs should take care of this feature it may be eliminated from the comparison of types of condensers or types of air pumps. It is conceded that the author's division of condensers and air pumps into good, indifferent and bad classes, in accordance with the vacuum and discharge-water temperature obtained, follows lines which have been generally accepted in the past; but a view from an engineering standpoint must consider the

impurities in the steam in the shape of air and non-condensable vapors, before judging any particular type of condenser or air pump.

The Author is exceedingly grateful for the interest shown in the paper and the practical nature of the discussion, which has served to clear up several ambiguities and to extend the subject into channels of inquiry representative of everyday commercial problems.

2 Mr. Ennis deprecates the loss by windage of considerable volumes of circulating water, in excess of that supplied by condensed steam. Theoretically, without windage loss, there should be practically no make-up water required, as an exact thermal balance has been established. But this loss does occur in both forced-draft and open tray type towers, and often to a serious extent. However this is simply a point in favor of the closed natural-draft type of tower, in which the velocities are reasonably low and hence small tendency exists to abstract water from the cycle.

3 The high loss in the Duquesne Lighting Co. Plant, it should be explained, is not due to windage. The hot jacket water can only be partially cooled, consequently enough must be thrown away to lower the temperature by the addition of fresh cold water. The loss at Potosina, however, was entirely due to windage.

4 The curves in Fig. 7 may very possibly be slightly in error, as they were necessarily based upon arbitrary assumptions—hence no attempt was made at absolute accuracy.

5 Mr. Foran evidently has had in mind the surface condenser in discussing possible and probable temperature differentials, whereas the author has referred more particularly to the barometric or jet types, especially in Fig. 7. This should have been stated more clearly in the paper. Generally speaking, it is possible with the jet type to work with much lower differentials than with the surface type. Mr. Foran's deductions regarding the extent of surface required to meet special conditions are therefore entirely proper. This very difficulty which is experienced with surface condensers in meeting the conditions imposed by the best cooling tower practice, only emphasizes in the author's mind the inherent advantages of the jet types.

6 The term "fixed cooling tower performance" could not apply to the construction of the curves in Fig. 7, as it is here used in the sense of efficiency rather than size. The use of "performance" here was in reference to relative cooling effect (deg. fahr.)—not capacity for absorbing heat—for the sake of eliminating another variable in the construction of Fig. 7. The size or capacity for a given con-

dition is simply a function of a heat quantity (B.t.u.) absorbed from the exhaust steam. For a given type of surface and draft velocity, the rate of absorption is fairly constant—a parallel to the constant rate of heat transmission through the tubes, as cited by Mr. Foran.

7 In reference to the Detroit tests, Table V, it should be noted that the condensing plant was not well adapted to the work in view, being an equipment temporarily retained in service from an old plant, too limited in surface and without means of operating air and water pumps individually, as required for economical working. The poor results from this particular plant were therefore distinctly attributable to the temporary nature of the installation, and not to an inherent fault in the type itself, as might be gathered from the reports.

8 In his closing remarks, Mr. Foran seems to confine the use of "natural-draft tower" to the open tray type. It is quite true that this has no application where large capacities or the highest efficiency are necessary. The closed chimney type is not dependent to any extent upon lateral wind velocity, and may be designed to economize space effectively.

9 The point raised by Mr. Dreyfus in regard to the effect of low temperature differentials is well taken. The author's observation that poor vacuum and good cooling go hand in hand applies to a given equipment, but the highly efficient condenser with low differential of course finds the most direct application.

10 The author did not observe or infer that the cooling tower field remains comparatively unexplored, but that certain conditions have tended to render the subject a closed book. This is not the case with engines, turbines, boilers, condensers, etc., so the fact that this condition obtains with cooling towers is not readily justifiable.

11 The two series of tests could not be presented in identical form, as the data were not available in such form. However, the curves, Figs. 11, 12 and 13, were drawn up to facilitate comparison. The first test covered day and peak loads only; the second, the entire 24 hours,—hence a low average load, as Mr. Longwell observes. Because the tower shows a low rate of heat dissipation with the entire surface installed, it should not be inferred that the actual work done was proportionately lower. Considering abscissae (B.t.u.) as equivalent to load (kw.) it must be apparent that for the same load a much higher cooling effect was obtained with the cooling surface complete.

12 For equal temperature heads, the cooling is bound to be the same except when the "lost head" differs, as it does slightly in Fig. 12. This opens up an extremely interesting line of inquiry—a survey

of rates of heat dissipation and humidity in each successive zone of the tower. Which part of the tower does the most work? Assuming air to be discharged exactly saturated at the temperature of exit, what spacing of mats is correct to produce a proper gradation of humidity from, say 70 per cent at entrance to 100 per cent at exit?

13 Regarding the inconsistency of Fig. 13, Mr. Longwell has forgotten to reckon the "lost head" shown in Fig. 12—approximately 40 deg. There is thus a very small discrepancy. However, it is hardly safe to interpolate in such a case. It is already pointed out in the paper that the tower is working at a disadvantage, owing to the extremely poor condenser performance, that imposes an extra burden on the prime mover as well.

14 The circulating water ratios adopted as a basis of the curves in Fig. 7, were so adopted to approximate average practice, otherwise a "family" of curves would replace each single curve shown.

15 Mr. Coffey favors the use of vapor pressure in lieu of relative humidity. The author entirely agrees to this method as more scientific. However, absolute humidity expressed in grams per cubic feet perhaps has a more direct bearing on cooling tower work.

16 The suggestion "to avoid free falling water" should have been amplified in the paper, and Mr. Coffey justly directs attention to it. Compactness or maximum duty for a given size is so essential in restricted locations that the atmospheric type is handicapped, if not debarred, which he himself recognizes in the closing sentence. The paper is directed entirely along these lines of maximum duty, and especially toward the development of the natural-draft type.

17 Mr. de Laval advances the argument that a tower should not have to be designed, rated and purchased *entirely* on a peak load basis. This is entirely in agreement with the author's object in presenting the combined natural-force-draft tower with fan auxiliary for use only during peak loads or during bad weather.

18 The objections of Mr. de Laval to the construction of the Butte tower are, however, not well taken, as the construction is more substantial than as described by him, and several years' service has not developed the defects he mentions.

19 The tests made at Detroit occurred, it is true, during the colder season, but in Par. 29 it is stated that the tower showed very little difference in operation in winter or summer—this on the advice of the chief operator.

20 Tables IV and V present the temperatures asked for to substantiate the assertion of a safe rate of heat dissipation of 200 B.t.u. per square foot per hour for the lath mat construction.

PUMP VALVES AND VALVE AREAS

By A. F. Nagle, Published in the Journal for Mid-October

ABSTRACT OF PAPER

This paper is designed to call the attention of engineers to the need of revising the common notion that "valve-seat area" is synonymous with "velocity of flow." It is evidently the purpose of specifications for pumping engines to secure a low velocity of flow through the valves, thus reducing the head required to force water through the pump; but to accomplish this laudable purpose, special and intelligent attention should be given to the *springs* of the valves, rather than to valve-seat areas. If that be done valve seat areas need not be greater than the plunger area for the vertical triple-expansion pumping engines so largely used in city pumps. A slight economy in both construction and operation could be effected by giving more study to the proper design and strength of springs for pump valves.

DISCUSSION

Charles A. Hague. The practice referred to by the author, of specifying that the area through the pump valves of waterworks engines shall bear a certain relation expressed in percentage of plunger area, is becoming less frequent, and it is to be hoped that it will finally be disregarded altogether. The relation between the plunger areas is merely incidental, because the valve area is a function of the quantity of water to be handled, the important matter being the velocity of the water through the valve seats to fill the plunger chamber as nearly complete as possible under the conditions.

2 The total valve area, or total area of valve-seat opening, ought to depend upon the velocity needed to pass the required quantity of water in a given time. Some authorities advocate a velocity not to exceed 3 ft. per sec., others 4 ft. per sec. and some as low as $2\frac{1}{2}$ ft. per sec. Two factors are to be considered, as follows:

3 First, as to the lift of the valves. The lower the pressure, and the lower the speed of the engine, the higher the valve may lift; on the contrary, the higher the pressure, and the higher the speed of the engine, the less the valve may lift, if a smooth, easy running, economical engine is desired.

4 Second, regarding the circumferential area of the valve space, or the area of the space around the edge of a disc valve, when it is open or off its seat. This is a factor that need not be very seriously considered, because the water, having succeeded in getting easily through the grating formed by the seat, will meet with very little resistance in moving out from under the valve. Valves free to lift to an unchecked height will often get so far away from their seats that slamming will take place at the reversal of the plunger. A pumping engine will work best when provided with sufficient valve-seat area to keep the mean velocity of the water down to about 3 ft. per sec., the lift of the valves being so restricted that they will return to the seats when the plunger approaches the end of its stroke.

5 With reference to plunger travel in conjunction with pump valve area, mentioned or inferred in the paper, the vital question is, How shall we obtain any certain plunger travel per minute: by a short stroke at many revolutions per minute, or, by a long stroke at few revolutions per minute?

6 After the water is well started through the pump valves, a larger increase in speed would be permissible than is found in practice, if it were not for the reversals at the end of the strokes. The 250-ft. per min. plunger travel mentioned in the paper, would be permissible with a 60-in. stroke at 25 r.p.m., or better with a 72-in. stroke at 21 r.p.m. The pump valves would work in a very satisfactory manner, the pumps would give very good hydraulic efficiency and the engines would run smoothly. But if we should attempt to obtain 250 ft. per min. with a 30-in. stroke at 50 r.p.m. there would be a great reduction in economy, smoothness of running and general efficiency.

7 The items in Par. 7 are all within the scope of mechanical efficiency, and will be reasonably well taken care of, if the valve factor is properly attended to. The most effective method for dealing with the question of valve area, is to establish a certain satisfactory area per unit of pumpage, at some definite minimum rate of revolution as a standard. Then, for every revolution per minute above the standard rate, add a certain per cent to the standard valve area. This will give an engine of more revolutions, a greater proportionate valve area than a slower machine, thus in fast engines keeping the valves nearer to their seats than in slow ones.

8 In Par. 26, the author makes a statement, with which one feels compelled to ask issue: "———the total valve area in this type of engine need not be more than the plunger area." As already pointed

out, there is no necessary relation between the valve and the plunger area at all. The relation is only incidental, or whatever it happens to be after the proper proportions are established. A certain area of plunger, with a certain stroke, at a given number of revolutions per minute sets up a certain velocity in the water through the valve seats. A plunger of half the area, with the same stroke and at twice the revolutions per minute, will set up the same velocity of displacement, and consequently the same mathematical velocity will be required through the valve seats; although the increased frequency of opening and closing will introduce another element for consideration, which will call for a greater proportionate valve area, for the greater number of revolutions per minute. In other words, a larger plunger running slowly will require the same valve area as a smaller plunger running faster, so far as the calculated displacement and velocity are concerned. The valve area in both cases depends upon the quantity of water and the selected velocity through the valve seats, regardless of the size and speed of the plungers.

9 The spring diagram and expressions are very nicely worked out, but the differentiation is too fine for real work, and could be mostly avoided by keeping the valves closer to their seats and avoiding refinement in springs. The idea is to get away from the laboratory engine, determine the conditions to be found in a pumping station, and then meet those conditions as they really exist, rather than try to adjust the working conditions to some real although impracticable refinement in some particular factor.

10 In many pumping engines now at work, some of the details worked out very nicely on the drawing board but failed to meet the actual requirements. There are waterworks engines of the cage pump construction, in which the ends of the valve stems, with valves exactly like those shown in the paper, have been sawed off, the valves being kept in place by means of wooden wedges, just because someone who never saw the inside pump after it left the shop, did not understand the requirements involved in the care and maintenance of the machine. In one or two such cases, the cages were difficult to remove, and there was not room enough to remove the valves, with the cages in the pump chamber, by the regular method of taking off the spring guard.

Irving H. Reynolds. Mr. Nagle calls attention to two very common errors which purchasers of pumping machinery fall into when preparing specifications:

a The absurdity of specifying the ratio between plunger and valve area without other limiting clauses.

b Specifying an unnecessarily large amount of valve area. Mr. Nagle suggests as a remedy for the first, specifying *velocity* through the valves rather than a *percentage* of plunger area, and for the second, the use of lighter springs, thus enabling the valves to rise to their full lift and thereby reduce the number of valves required.

2 In regard to the first, there is an increasing tendency among engineers to specify a maximum velocity of flow through the valves rather than their area relative to the plunger.

3 Quietness of operation rather than cost is the first consideration in the design of pump valves, and the present excessive valve areas have grown from this idea. Time is also an important element in determining pump valve action; therefore, the number of reversals or valve seatings, rather than the piston speed, is the important factor, and consequently valves of small diameter and therefore of relatively low lift, have displaced the large diameters in common use a few years ago.

4 To further decrease the lift of the valves and, therefore, permit them to close quickly and quietly at high speeds, valve areas have been increased to a point where in actual operation the valves lift only a fraction of the theoretical height to which they should lift to give a full opening; in other words, large valve area is provided for the purpose of not using it.

5 If on a high-speed (high-revolution) pump the valves were fitted with light springs, permitting them to lift to their full height as suggested by Mr. Nagle, it is probable that the pump would be exceedingly noisy, as the valves would be so far from their seats at the time of plunger reversal that they would not seat until the flow through them had reversed, and this slowness in seating would be still further aggravated by the light springs employed. There is no doubt, however, that in many cases the springs used are unnecessarily stiff and on slow-speed engines the lighter springs would be found satisfactory.

6 In earlier practice, particularly with direct-acting pumps, the valve area was small in proportion to the plunger and the valves were obliged to lift nearly to their full height. In this type of pump, as the plunger speed was relatively high to nearly the end of the stroke, the valves became noisy if the pumps were operated at high speed.

7 With the general introduction of the crank and flywheel pump came higher rotative speeds and the necessity for larger valve area and smaller valve opening, i. e. lower lift, until present practice has crystallized at velocities of 3 ft. to 3½ ft. per second through the valves, and valves of between 3½ and 4 in. in diameter for ordinary waterworks service. In general, the best results would be obtained if engineers in drawing specifications would limit the mean velocity of water through the valves at about 3 ft. per second and the diameter of the valves to not over 4 in.

F. W. Salmon. I prefer to make these valves somewhat different from the one illustrated in the paper. I do not believe it is best to use the radical ribs of the valve seat to screw it in, but that it is better to cast small projections on the outside, as at *A* Fig 1. This part is of such a size that an ordinary black pipe will fit neatly when properly milled out at the end, thus making a good socket wrench at a minimum cost.

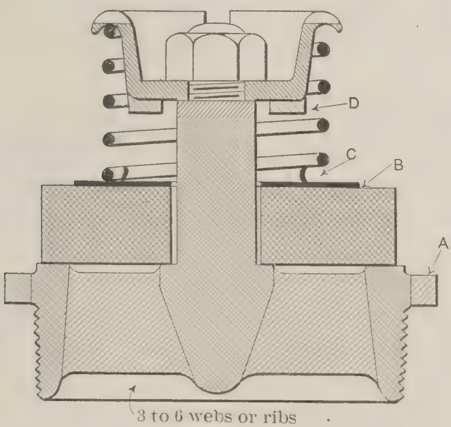


Fig. 1 Cross Section of Pump Valve, Showing Improvements Suggested by Mr. Salmon

2 I prefer to put a brass plate on the top of the rubber valve, as shown at *B* Fig. 1, and to partially punch out and turn up little projections from this plate as at *C* Fig. 1 and Fig. 2. The plate prevents the spring wearing into the top surface of the valve, and the projections keep the spring properly centered.

3 Small projections should be cast on the under side of the spring guard as shown at *D* Fig. 1 and Fig. 3, the latter being the under side

of the spring guard. If the valve is ever drawn so high as to come into contact with these projections it will still descend freely, not being in the least hindered by the soft surface of the valve forming a close contact with the smooth under surface of the spring guard, as it

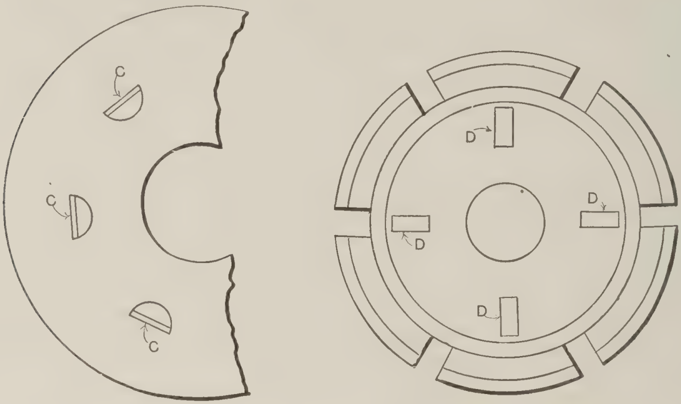


Fig 2 and 3 Showing Projections on Brass Valve Plate and on Spring Guard

is sometimes made. I consider that this is useful in cases of fast running pumps, as in such machines it is particularly desirable to have the valves seat while the crank is passing the dead center, and so a quick closing action is required.

William Kent. I hope Mr. Nagle will supplement the paper by telling us what proportion of valves and valve springs he would use for certain conditions. The paper is now largely one of criticism, and I would like to have the author make it a constructive paper. Par. 25 reads "The place to begin the study of proportions of a pump is at the spring of the valve. Make a sample spring of such diameter and length and strength as you may think desirable, and by experiment construct a diagram of its rate of compression, as in Fig. 1." This is good advice for pump designers, but other mechanical engineers are called in to confer about these points, and if Mr. Nagle would tabulate the proportions of springs suitable for pumps, and give the lifts at certain velocities of water, his paper would be more useful to these engineers.

2 The author criticises the practice of specifying the percentages of area of the valve and the pump. I see nothing very wrong

in that, provided the plunger area and the speed are also specified, as is usually done, otherwise some of the bidders will put in a small pump. In order to compel them to supply a pump large enough, we limit the velocity of the plunger; and having limited the velocity of the plunger and specified its size, we may as well say that the valve must be so many per cent of the plunger area, as to state what the velocity of the water must be. The specification is good enough, provided these additional items of plunger area and speed are also specified.

Prof. R. C. Carpenter. It is quite evident to any one familiar with hydraulics that the difficulties from the narrowing of the valve are largely inherent in the spring. If a spring could be obtained which would open uniformly with increase of pressure the troubles due to certain inertia effects which are mentioned, would disappear. This, however, merely points out the source of trouble and leaves the question open as to what shall be done. In substance, defects are merely pointed out without remedies. I would suggest, if Mr. Nagle can, that he give some of these remedies for the troubles which he has described.

E. H. Foster. Attention should be called to the fact that this paper refers to the valves of one type of pump. Many pumps of other types are built, particularly those without fly wheels, to which it is not absolutely necessary that these rules should apply. It is well known that a considerable pause at the end of the stroke of the duplex pump facilitates the closing of the valves, so that these empirical rules for lift and area must be quite different for that type of pump.

The Author. Some new matter which has come to the attention of the writer of this paper, is appended herewith. A careful study of this will answer most of the points raised in the discussion, especially the point made by Mr. Reynolds and Mr. Hague, to the effect that the maximum velocity through the valve should be limited to 3 or 4 ft. per sec. It can be assured that the formula and Table 1, quoted from Professor Bach's experiments governing the relation of spring pressure and velocity of flow to the coefficient of contraction, is correct.

TABLE I PROFESSOR BACH'S EXPERIMENTS WITH A FLAT VALVE AND A FLAT SEAT (SEE FIG. 1)

Inside diameter of valve seat d = 1.968 in. Outside diameter of valve d_1 = 2.362. Ratio of inside and outside areas, 1 to 1.44. Inside area, 3.04 sq. in.

		H = 1.27 − 1.29 ft.				H = 3.08 − 3.11 ft.	
1	M = lift of Valve, in.........	0.23	0.55	1.01	0.122	0.40	0.65
2	G = Weight of valve, lb......	2.028	2.218	2.304	4.610	5.073	5.238
3	Q = Volume of water, lb. per sec.........................		6.548	8.554	3.086	6.768	11.20
4	H = Head of water, ft........	1.29	1.29	1.27	3.11	3.10	3.08
5	W = Velocity through seat, ft. per sec..................................		4.97	6.46	2.33	6.17	8.46

Calculations by Nagle

6	$\frac{m}{d}$ = Ratio lift, to diameter.	0.12	0.28	0.51	0.06	0.20	0.33
7	G = p Weight per sq. in., lb. per sq. in...........	0.666	0.728	0.760	1.516	1.668	1.723
8	V_g = Velocity due to p, ft. per sec...............	9.55	9.47	9 07	14.70	14.72	14.35
9	V_h = Velocity due to H, ft. per sec................	9.12	9.12	3.92	14.14	14.10	14.07
10	Ratio of $\frac{V_g}{V_h}$..................	1.04	1.04	1.02	1.04	1.04	1.02

Line 7 is obtained by dividing the weight G, given in line 2, by the area of d, or 3.04 sq. in.

Line 8 is obtained by the aid of Table 2, where opposite the value of $\frac{m}{d}$ is found the coefficient and formula. For example: taking the first case of a lift of 0.23 in., or a percentage of 0.12 of the diameter 1.968 in., we find by interpolation in Table 2, the formula, $V = 1.17\sqrt{100\,p}$, or $V = 1.17 \times 8.16 = 9.55$ ft. per sec.

Line 9 is obtained from the fundamental hydraulic formula $V = 8.025\sqrt{H}$, when H is the head in feet and V the velocity in feet per second. For example, in the first case cited we have, H = 1.27 ft. $V = 8.025\sqrt{1.27}$, or = 9.12 ft. per sec.

Line 10 is self-explanatory, and is introduced as a check upon the work and formulæ, as if correct, it should be unity. The slight deviations are due to the various decimals not being carried far enough, but they are carried far enough for all practical purposes.

TABLE 2

1	$\frac{m}{d}$ =	....	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.5d
2	u =	....	0.65	0.60	0.56	0.53	0.50	0.47	0.44	0.41	0.37
5	p =	0.67	0.69	0.72	0.74	0.77	0.80	0.83	0.86	0.89	$0.92\frac{V^2}{100}$
6	v =	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	$1.04\sqrt{100p}$

Line 1 $\frac{m}{d}$ is the actual rise, or lift, of the valve, divided by its inside seat diameter.

Line 2 u is the coefficient of contraction at the point of discharge with a given lift.

Line 4 V is is the velocity of the issuing stream at the point of discharge in feet per second.

Line 5 p is the pressure in pounds per square inch and is found by dividing the weight of the valve (in water), plus its spring pressure in pounds by its inside seat area in square inches.

Line 6 v **is the velocity of the issuing stream per second.**

2 Let us apply the formula to the 3 ft. per sec. assumption. For a lift of, say, 0.20 × diameter,

$$p = 0.77 \times \frac{V^2}{100} \text{ or } = 0.07 \text{ lb. per sq. in.}$$

of inside seat area. Such small spring pressure is out of all proportion to what common practice has established, which is from 0.30 to 0.60 lb. at the initial and 0.75 to 1.50 lb. at the full lift. The formula for the resulting velocities is very simple. Suppose we solve for four spring pressures of, say, 1.50, 1.25, 1.00 and 0.85 lb. at full lift, and 40 per cent, or 0.60, 0.50, 0.40 and 0.34 lb. at the initial point. at a lift of 0.20 × diameter, the formula would be

$$V = 1.14 \sqrt{100 \times p}$$

and the velocities for

$$\begin{aligned} p = 1.50 \text{ lb.} \quad & v = 13.96 \text{ ft. per sec.} \\ 1.25 \text{ lb.} \quad & v = 12.74 \text{ ft. per sec.} \\ 1.00 \text{ lb.} \quad & v = 11.40 \text{ ft. per sec.} \\ 0.85 \text{ lb.} \quad & v = 10.51 \text{ ft. per sec.} \end{aligned}$$

The coefficient of contraction would be 53 per cent in each case.

3 It is plain, therefore, that we are far from realizing four feet per second with our present spring practice.

4 To Mr. Reynolds: The writer did not mean to lighten the springs abnormally, in fact, 0.45 lb. to 0.50 lb. initial is probably light enough, but if they could be made somewhat longer, so as not to tighten up too rapidly, it would seem to be desirable.

5 To Mr. Kent: The formulæ given by Professor Bach are a very great addition to our knowledge of pump-valve action. Within the limits prescribed, we have now a safe guide for valve construction. What it should be for other numbers of revolutions and plunger velocities, I am not able to formulate. Professor Haeder goes into that phase of the problem, but as his theory is not confirmed by extensive experience, I do not take it up in this paper.

ADDENDUM TO PAPER

6 In Par. 15 of the paper is given a formula for ascertaining the lift of a pump valve, from which was omitted, as stated, the coefficient of contraction. Not knowing the value of this coefficient with certainty, the writer hoped the information would be supplied in the dis-

cussion. The omission was not referred to, however, and he is now able to supply it himself.

7 In a German book on pumps and pump valves by Herm. Haeder, Duisburg, the subject is treated in an exhaustive manner. The actual coefficients of contraction are given, with the results of impact upon the valve, based upon experiments by Professor Bach. In what follows reference is made only to that part which bears on the subjects of flat valves and flat seats, of which the inside and outside areas bear the ratio of 1.00 to 1.44. The notations were originally in French, but in what follows have been transformed into English units.[1]

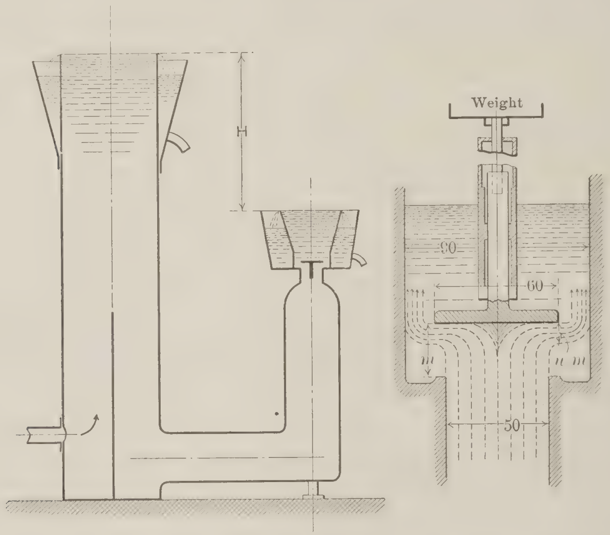


Fig. 1 Apparatus used by Professor Bach for Determining Coeffcient of Contraction

8 Fig. 1 shows the apparatus used by Professor Bach. Table 2 (Haeder 261) gives the original data and results obtained and some calculations of my own, the better to elucidate the subject.

9 Fig. 2 and Fig. 3 (Haeder 110 and 110a) show a valve closed and one open, with the respective formulæ for the two positions of the valve, giving the values for velocity or pressure in the two extreme positions. "Open" signifies a lift of one-half the diameter, which, needless to say, is far beyond American waterworks practice.

[1] The original tables in French units can be referred to in the author's manuscript on file in the rooms of the Society.—Editor.

10 Table 2 (Haeder 213) gives the values of v and p for the intermediate positions of the valve, and also the value of "u", the all-important coefficient of contraction at all positions. Use this table to ascertain the velocity v of the water through the valve opening and also the coefficient of contraction u at the same point.

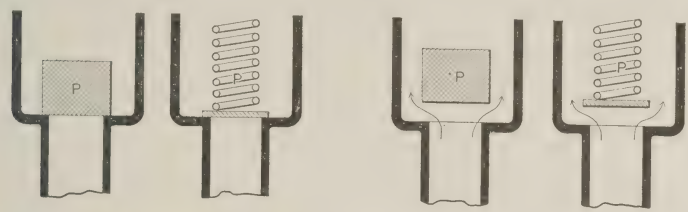


FIG. 2 AND FIG. 3 SHOWING, RESPECTIVELY, A VALVE OPEN AND A VALVE CLOSED. THE FORMULAE FOR THESE TWO POSITIONS ARE AS FOLLOWS:

V = VELOCITY IN FEET PER SECOND P = POUNDS PER SQUARE INCH

VALVE CLOSED	VALVE OPEN
$V = 1.22\sqrt{100\,P}$	$V = 1.04\sqrt{100\,P}$
$P = 0.67\,\frac{V^2}{100}$	$P = 0.92\,\frac{V^2}{100}$

11 We can now say that we have a practically correct formula for ascertaining the volume of water discharged through a flat disc pump valve of a certain diameter, an assumed lift, and a certain tension of spring.

12 Throughout all the following calculations a maximum lift of valve of 0.15 d, is taken, leaving the reader to make for himself other assumptions of lift and the consequent calculations. Various tensions of springs will be taken, to illustrate the importance of giving more attention than heretofore to the strength and length of springs.

13 Take, for examples, the same dimensions of pump and valve as those used in Par. 16. Formula 3 (Par. 15) would now be better expressed in terms of N, the number of valves, than assuming the number of valves and solving for the lift L. The formula would then read,

$$N = \frac{P \times V_m}{C \times L \times u \times V_m}.$$

Applying this formula to the three different strengths of springs before used, we get the following results:

14 First, ascertaining the velocity through the valve by the aid of Table 2, the spring tensions were as follows;

Case 1: initial, 0.60 lb.; final 1.55 lb. per sq. in.
Case 2: initial, 0.40 lb.; final 1.03 lb. per sq. in.
Case 3: initial, 0.30 lb.; final 0.77 lb. per sq. in.

The formula for the velocities due to these final pressures at a lift of 0.15 d, are

$$\text{Case 1: } V = 1.16\sqrt{100 \times 1.55}, \text{ or } V = 14.41 \text{ ft. per sec.}$$

$$\text{Case 2: } V = 1.16\sqrt{100 \times 1.03}, \text{ or } V = 11.77 \text{ ft. per sec.}$$

$$\text{Case 3: } V = 1.16\sqrt{100 \times 0.77}, \text{ or } V = 10.18 \text{ ft. per sec.}$$

The coefficient of contraction in each case is $u = 0.56$. Substituting these values in Formula 4, we have,

$$\text{Case 1: } N = \frac{908 \times 6.67 = 6056}{10.53 \times 5.625 \times 0.56 \times 14.44} = 127$$

$$\text{Case 2: } N = \frac{6056}{3.317 \times 11.77} = 155$$

$$\text{Case 3: } N = \frac{6056}{3.317 \times 10.18} = 180$$

15 Let us make a similar calculation for springs of the same initial strength, but longer, so that they will tighten only one-half as much in their nine-sixteenths lift. Then the first spring final tension becomes 1.08 lb., the second spring 0.72 lb., and the third spring 0.53 lb.; and the velocities become

$$\text{Case 1: } V = 1.16\sqrt{100 \times 1.08} = 12.05 \text{ ft. per sec.}$$

$$\text{Case 2: } V = 1.16\sqrt{100 \times 0.72} = 9.84 \text{ ft. per sec.}$$

$$\text{Case 3: } V = 1.16\sqrt{100 \times 0.56} = 8.68 \text{ ft. per sec.}$$

and solving for N in formula 4, we have

$$\text{Case 1: } N = \frac{6056}{3.317 \times 12.05} = 152$$

$$\text{Case 2: } N = \frac{6056}{3.317 \times 9.84} = 186$$

$$\text{Case 3: } N = \frac{6056}{3.317 \times 8.68} = 210$$

16 To calculate the loss of efficiency for these different springs let us take the mean pressure on the springs to be the initial, plus two-thirds of the increase, and twice this for the two strokes; and this sum must be divided by the total pump head, say 80 lb., to obtain the loss of efficiency. We would then have

Case 1: $[0.60 + (1.55 - 0.60)\ \frac{2}{3}] \times 2 \div 80 = 3.06$ per cent
Case 2: $[0.40 + (1.03 - 0.40)\ \frac{2}{3}] \times 2 \div 80 = 2.05$ per cent
Case 3: $[0.30 + (0.77 - 0.30)\ \frac{2}{3}] \times 2 \div 80 = 1.50$ per cent

With stronger springs, we would have

Case 4: $[0.60 + (1.08 - 0.60)\ \frac{2}{3}] \times 2 \div 80 = 2.30$ per cent
Case 5: $[0.40 + (0.72 - 0.40)\ \frac{2}{3}] \times 2 \div 80 = 1.50$ per cent
Case 6: $[0.30 + (0.53 - 0.30)\ \frac{2}{3}] \times 2 \div 80 = 1.15$ per cent

Grouping these figures for better comparison, we have Table 3.

17 We have now, in Table 3, figures which enable us to study pump valve constructions in an intelligent manner. The formulæ given enable us to construct a similar table for any other assumed dimension of plunger and its velocity, height of lift of valve, or spring tension.

18 In conclusion the writer wishes to say that now, for the first time in the history of the modern high-duty pumping engine, we have a formula for designing a pump valve that is scientifically correct, and one based upon hydraulic experiments carefully made by a competent authority. The subject seems important enough to bear repetition in grouping the previous instructions, as follows:

19 Find the area of the plunger in square inches, and the maximum speed of the plunger in feet per second. The latter is found by multiplying the stroke in feet by the maximum number of revolutions per minute, multiplying this result by 1.60, to reduce it to its maximum velocity (the crank velocity), and dividing by 60 to reduce it to feet per second. This product, algebraically expressed by $P \times V_m$ in Formula 4, becomes the numerator of the equation.

20 Determine the size of the pump valve-seat and its net area between the ribs, whether the valve bears on the ribs or not; that will be the inside area of the valve against which the impinging stream acts.

21 Decide what lift of valve you intend to have. American water works practice is from 0.10 to 0.20, the diameter of the inside of the outer seat. This lift is designated by L in the formula.

22 Decide what spring pressure you will have, both at the beginning and at the full lift. This spring pressure is expressed in pounds

per square inch of inside valve area and usually runs from 0.30 to 0.60 lb. per square inch at the beginning of the lift, and it ought not to be quite double this amount when the valve is full open to its stop. It will be this final tension, plus the weight of the valve in water, designated by p in Table 2, that will be the determining factor for the velocity of the issuing stream. To illustrate, if the final pressure be 0.81 lb. per sq. in., with a lift of 0.15 d, the equation (see Table 2) for $V_m = 1.161\sqrt{81} = 10.44$ ft. per sec.

23 The discharging area is the net circumference of the inside valve diameter C, taking out the ribs whether they support the valve or not, multiplying this by the actual lift and this product by the coefficient of contraction u, as found also in Table 2, which, for the lift cited, is 0.56.

24 Algebraically expressed, these factors become the denominator in the Formula 4,

$$N = \frac{P \times V_m}{C \times L \times u \times V_m}$$

TABLE 3 $\frac{9}{16}$ IN. LIFT

PLUNGER 34 IN. DIAMETER BY 5-FT. STROKE BY 25 R.P.M. MAXIMUM VELOCITY = 6.67 FT. PER. SEC. VALVES $3\frac{3}{4}$ IN. INSIDE DIAMETER. NUMBER OF VALVES, SPRING TENSIONS AND PUMP EFFICIENCIES.

	Initial and Final Spring Tension Pounds	Valve Seat Area Per Ct.	Number of valves	Lift of valves Inches	Maximum Velocities Feet per Second			Loss of Efficiency Per Ct.
					Plunger	Valve Seat	Valve	
1	2	3	4	5	6	7	8	9
1..........	0.60 to 1.55	112	127	$\frac{9}{16}$	6.67	5.96	9.44 – 14.44	3.08
2..........	0.40 to 1.03	137	155	$\frac{9}{16}$	6.67	4.87	7.71 – 11.77	2.05
3..........	0.30 to 0.77	159	180	$\frac{9}{16}$	6.67	4.20	6.68 – 10.18	1.50
				Longer Springs				
4..........	0.60 to 1.08	134	152	$\frac{9}{16}$	6.67	4.98	9.44 – 12.05	2.30
5..........	0.40 to 0.72	164	186	$\frac{9}{16}$	6.67	4.07	7.71 – 9.84	1.52
6..........	0.30 to 0.53	185	210	$\frac{9}{16}$	6.67	3.60	6.68 – 8.68	1.15

Column 3 is obtained by multiplying the number of valves by the net area through the seat (8.00 sq. in.), and finding its ratio to the plunger area.

Column 5 is taken at $\frac{9}{16}$ in. = $(0.15 \times d)$ in all cases.

All the other data have been already explained.

AN EXPERIENCE WITH LEAKY VERTICAL FIRE-TUBE BOILERS

By F. W. Dean, Published in The Journal for October

ABSTRACT OF PAPER

This paper discusses the difficulties experienced with some large vertical boilers, somewhat over 10 ft. in diameter, and containing over 6000 sq. ft. of heating surface. The boilers leaked very badly very soon after being started and nothing that was done improved their condition until the water legs were lengthened from 2 ft. to 7 ft. $2\frac{3}{4}$ in. The boilers were raised 5 ft. $2\frac{3}{4}$ in. Before they were raised the lower ends of the tubes would cover with very hard clinker and become stopped up. This clinker could be removed only by cutting it off when the boilers were cold. After the boilers were raised, a light clinker that could be blown off formed about the tubes; by removing this by blowing every three or four hours the leaks were stopped and they have never returned.

The trouble varied with different kinds of coal Each boiler had been run constantly at over 1000-h.p. and the economy seemed to be about the same no matter what the power was. So far it has been difficult to obtain good combustion, but the heat-absorbing power of the boilers is admirable. The experience with these boilers indicates that there is no ordinary limit to the size of a vertical fire-tube boiler.

DISCUSSION

Reginald P. Bolton. It appears to me that this design of boiler was an invitation to the trouble that followed, and it is only necessary to go back into the experience of other people to find out that others have suffered in the same manner. If the view of the boiler as presented in the paper is turned horizontally, and it is imagined that it is a locomotive boiler cut off short, it will be seen that there is no combustion chamber whatever in it. This boiler was to be put to a service which might call for a rate of combustion in the furnace which would demand double its rated capacity output, so that the double aggravation of a very small combustion chamber and very large rate of combustion, was present.

2 The design of the boiler is radically defective in two important points, namely, the tubes are entirely too long, and the combustion space was entirely too small. It is now very nearly half a century ago that the experiments of Geoffroy and Petiet demonstrated the futility of unduly lengthening fire tubes. These experi-

ments demonstrated the rapid reduction in efficiency due to length of tubes, under various conditions of draft and rates of fuel consumption. Almost precisely the same conditions were tested as in the author's boiler, as follows:

3 A consumption of fuel exceeding 50 lb. per square foot of grate, under an air pressure of 2.36 in. with the following results:

	Evaporation per Sq. Ft. Lb.
Fire-box plate	23.5
First three feet of tubes	5.4
Second " " "	2.5
Third " " "	1.33
Fourth " " "	0.83
Fifth, three feet evaporated only	0.48
Sixth " " " "	0.3

The last two were found by extending the curve.

4 An examination of these results might have dissuaded the author from the mistake of designing the boiler with such a length of tube, involving not only inefficiency, but the evident concomitant of leakage as a result of expansion and contraction. Apart from the other defective feature, the boiler could have been shortened so as to reduce the tubes at least five feet in length, and would no doubt have given better efficiency as a result.

5 The general type of the boiler possesses nothing new or original unless we may so regard the restricted combustion chamber, by which the tube plate was brought within seven feet of the grate, allowing a total capacity of only 535 cubic feet for the fire and for the gases of combustion.

6 A very simple computation of the results of the combustion of 40 lb. of coal per square foot of grate area, will show that the volume of products of combustion would be so great, that only an excessively heavy draft could force them through the combustion chamber and tubes, and that incomplete combustion was bound to result.

7 The addition of 5½ ft. to the height of the chamber, which was arrived at only after three years experience with this boiler, nearly doubled the effective space for combustion, and also removed the ends of the tubes from the direct action of the blast. It may be observed that a Dutch oven would have afforded equal results, at perhaps less expense.

8 The reason for the adhesion of molten clinker to the ends of the tubes, need have presented little difficulty, in the light of past

experience, since the ends of the tubes were placed so close to the fire. This result developed in the fire-tube boilers of H. M. S. Polyphemus nearly thirty years ago, and when found in the boilers of locomotives is due to precisely the same cause.

9 It will be noticed that the best of the tests which were made after the change of combustion chamber was effected, is that in which the rate of fuel consumption is least.

10 I agree with the quoted conclusion of the second boiler expert, referred to in Par. 6, and am at a loss to understand why such an opinion, thus expressed, was regarded as unsatisfactory. It may be hoped that the paper may stand as a warning signal to other designers. It requires a great deal of courage to present a paper of this kind, and the author should be thanked for bringing forward a record of a failure so that we may profit by the facts.

William Kent. I join with Mr. Bolton in praising Mr. Dean's courage in bringing forward a report of his failure, and I regret that some eight or ten years ago I did not bring forward a record of another similar failure, not my own, but that of some other man, which might have prevented Mr. Dean's bringing forward a record of this failure. The New York Steam Company bought a boiler for their Greenwich Street Station to go in a very small ground space. It was a very large plain vertical cylinder boiler, eight or ten feet in diameter, full of tubes about 20 ft. long, and was rated at 1000 h.p. It had not been in use more than a week or two when it began to leak. There was no way to clean the flat tube sheet or to clean the tubes of scale, and the boiler was condemned and taken out.

J. C. Parker. The reason that the tubes leaked was that when the boiler was set close to the grate the tube ends were subjected to wide fluctuations in temperature. The flow of air through a chain grate increases toward the rear end, and where the boiler was set higher there was more mixing of the hot and cold currents and, consequently, less fluctuation in temperature.

2 The clinkering of the tubes would naturally increase the trouble because of the concentration and increased friction of the gases in the tubes that remained clear.

Orosco C. Woolson. This discussion has brought out the important fact that perfect combustion should take place before the gases reach the tubes or shell of the boiler.

2 I have been somewhat surprised in my travels among the cotton and woolen mills of the Eastern States where the management have large experience in cotton spinning but are limited in personal experience regarding what constitutes the production of the highest calorific value of a pound of bituminous coal. One man of large experience in mill work wanted his furnace fire directly under the tubes of his vertical boilers, and gave me his reasons. I told him that I would guarantee him better results if he would discard the idea that the area immediately under and against the tube sheet should act as a combustion chamber. Let combustion take place entirely before it reaches the tube sheet and the results will be much more satisfactory.

3 Secondly, as to the tubes filling with vitrified slag or any other residuum of combustion, I would suggest that such deposit should be made to take place under a fire arch, where it will adhere to the crown and serve a useful purpose by forming a refractory coating. This practice is becoming popular, and more so today than ever before. It is my opinion that by completing combustion under a properly constructed arch within a properly constructed combustion chamber, the products of this combustion will be sent to the boiler in the form of what we will term "caloric ether" and not a mixture of its original constituents which play no useful part, under the circumstances, in producing or maintaining heat, but rather are subject to ready condensation.

A. A. Cary. In my experience with vertical fire-tube boilers I once found a boiler containing shorter tubes and of a greater diameter than are ordinarily found in the Manning type. The fuel used was a moist anthracite coal, and there was a natural draft of more than one inch of water in the smoke box over the boiler. The draft could not be regulated, due to the previous burning out of the steel plate butterfly damper. The partially burned furnace gases passed rapidly through the vertical tubes and ignited above the top tube sheet, thus causing the destruction of dampers and the steel breeching, to say nothing of the reduced evaporation in the boiler due to this waste of heat.

2 The trouble was remedied by placing the grates at a greater distance from the lower tube sheet and arranging baffles in the combustion chamber so as to insure the more complete combustion of the gases before they entered the tubes. A cast-iron plate damper replaced the former one of steel plate, and no further trouble has since been experienced.

3 In another case, the question came up as to the advisability of applying a special automatic furnace, using bituminous coal and producing very high temperatures, under boilers of the Manning type. An arrangement which has been used in New York City for a number of years was suggested and successfully applied.

4 Fire-bricks, piled on edge with open spaces between the bricks, were arranged a short distance beneath the lower tube sheet. This checker work of bricks filled the entire space beneath the boiler, the openings between the bricks at the center being very much reduced, so as to cause a decreased flow of gases directly under the center of the overhead tube sheet. By this means, a very even distribution of temperature was secured over the entire area of the lower tube sheet with a slight reduction of heat delivery at its center, the most sensitive portion of the whole tube area.

5 The author mentions inefficient combustion, which is indicated by the comparatively low percentage of CO_2 and high percentage of O, shown in Table 1. As the higher temperatures are secured by the most complete combustion with the least excess of air, the question arises, why should such destructive results follow such inefficient furnace conditions?

6 Pyrometric testing with gas analyses have taught me that when a furnace is being operated inefficiently, very high temperature may be found in one part of the furnace while at the same time a comparatively low temperature may exist in another part. This may lead to the simultaneous impingement of gases of very different temperatures upon various parts of the lower tube sheet, setting up destructive strains and contributing to such troubles as have been described by Mr. Dean.

7 The lower tube sheets of boilers of the Manning type are very sensitive, especially towards the center of the sheet where the water seems to penetrate with great difficulty, thereby failing to keep this portion of the heating surface constantly wet.

8 Concentration of heat due to concentration of combustion and lack of space for this small volume of high-temperature gas to diffuse itself throughout the entire mass of furnace gases before they reach the tube sheet, is bound to cause trouble, especially when this highest temperature is concentrated against the center of the tube sheet on the inner surface of which there is apt to be little or no water. After the center of this sheet loses the supporting effect of the center tubes, acting as stays, the surrounding tubes are very apt to follow.

9 Concerning the low efficiency of the furnace referred to in

Par. 13, there should be no trouble in remedying this fault. A properly conducted furnace test (apart from the boiler) with pyrometers, gas analyzing apparatus, etc., will show just where the trouble exists and will point out the needed changes as well as the limitations under which this type of stoker can be worked with the different grades of fuel used.

Prof. L. P. Breckenridge. One of the speakers said that the highest temperature in a boiler furnace is directly over the fire. This is not always so. We have measured the temperature twenty feet from the fire and found it higher. It depends on the volatile content of the fuel and whether the flame has been supplied with a sufficient amount of air early in the process of combustion. It is this that determines whether the high temperature point is ten feet or twenty feet away. Many times in our experiments in the St. Louis boiler trials we have seen that every time the furnace door was opened the temperature at the rear end of the combustion chamber went up, because when more air was admitted the combustion was better and the temperature increased.

2 For experiments concerning the transmission of heat through a boiler tube, it occurs to me that Mr. Dean has designed one of the most satisfactory laboratory boilers I have seen. There has been much discussion of late on the heat transferred through a boiler tube, as influenced by the velocity of the gases passing through the tube. This boiler with its large number of tubes would be just the type with which to make a test on this point. I wish Mr. Dean would burn a large amount of coal per square foot of grate in this boiler furnace, using, first, all the tubes, and secondly, only one-half the tubes. If the same amount of coal was burned in each case the velocity of the gases through the tubes would be twice as great in the second case, and it would be interesting to know the relative amounts of heat transferred.

3 I hope that some time we may take up the question of the burning of fuel, making a distinction between the economical performance of the boiler and of the furnace. We have reached a time when we can intelligently discuss these questions separately. Anthracite coal, on account of its high fixed-carbon content, is burned mostly on the grate itself. When burning semi-bituminous coal, with 18 to 20 per cent volatile content, a large combustion chamber is required, and as the volatile content increases the size of the combustion chamber must be increased. When burning bituminous coal, with

40 per cent volatile content and 20 per cent ash, the fuel actually burned on the grate is small. The grate supports the fuel and some coal is burned there, but it is in the combustion chamber that we burn fully one-half the combustible part of our fuel. It is evident that more attention must be given the proportions of our combustion chambers when burning high-volatile coals, and especially at high rates of combustion.

Prof. A. M. Greene, Jr. In London Engineering for October 22 and November 5, 1909, appeared an article by Professor Dalby, in which he summarized a number of articles referring to heat transference through plates. I would commend the article to the attention of all the members of the Society interested in this matter.

2 In London Engineering for Feb. 1909, Professor Nicholson described experiments showing clearly that only a small part of the possible heat transmission through plates is utilized. I mention this to call attention of the members to the fact that some data are available on this subject. In this article are given the formulæ for heat transmission which may by compared with the results of German Experiments recently completed at Dresden (Zeit. des. Verein Deutscher Ing., October 23, 1909).

William Kent. In another issue of London Engineering, a correspondent showed that the idea of high speed of the gases being favorable to combustion was negatived by the Lancashire boiler, in which the flues are very large and the speed of the gases low, yet the economy is as high as in any other boiler.

Reginald P. Bolton. It is mainly a question of the difference in temperature between the inside and outside of the heating surfaces. The lower the temperature of the feed water, and the higher the temperature of the fire, the greater will be the efficiency of the boiler.

E. D. Meier. I find myself in substantial agreement on some points with all the gentlemen who have spoken. I want to say for Mr. Dean, that he is correct in his conclusion that the precipitation which occurs at the bottom of the tubes has a great deal of influence on the overheating of the tube sheet. The other causes which were mentioned are also true, but there is no doubt an accumulation of carbon there. I do not know whether Mr. Dean preserved any of the

precipitate or stalactites, but I believe a large part of it was unconsumed carbon, which will remain at a high temperature for some time.

2 I am reminded of an experience which I had with water-tube boilers at the Chicago World's Fair. I think there were ten different makes of water-tube boilers, most of them sub-horizontal, but some of the vertical-tube type and some of the bent-tube type. We were burning crude oil, and all the boilers suffered from the same causes,—every one lost tubes by burning out. Some were careful enough to shut down a boiler as soon as they noticed the blisters on the tubes.

3 The boilers which I had at Chicago were afterward placed in the midwinter fair at San Francisco, and were fired with California crude oil for seven months without a tube being lost. These boilers were afterwards sold with the condition that if the customer found any tube damaged it would be replaced, but not one was found to be burned. That bears on the subject mentioned by Mr. Dean. The trouble we found was this: The oil is supposed to be atomized in the burners, but this is not always the case. Little slugs of oil would fly up and adhere to the tube, and would spread and slowly carbonize. They would not burn, because no air could get to them. One little spot, a half inch in diameter, would become red hot in spite of all the circulation of water, and would ultimately burn out and make a blister.

4 When the boilers were installed in California, the oil burners were placed lower and were directed downward so that the jet would strike the bottom of the combustion chamber at a distance of six feet from the front, hence there was no chance of oil striking the tubes. Perfect combustion was obtained, and on one occasion one of the boilers was forced so hard that a picture was taken of the inside of the furnace by its own heat. I have that photograph still, to show what can be done. One can see a perfectly white heat and not a single blister on the tube. In Mr. Dean's case carbon was deposited and became incandescent, and gave an intense local heat on some point, which accounts for the failure of the tubes at such point.

5 In regard to the combustion chamber, I agree with Professor Breckenridge. I have always been a believer in a large combustion chamber, and one of my early experiences in that direction was when in charge of a plant having two horizontal tubular boilers, using Illinois coal. At that time everybody in the Mississippi Valley believed in river practice. The boilers, engines and dimensions of pipes, etc., were according to river practice. The boilers were set with the grate twelve inches from the bottom of the shell. I

raised them to thirty inches, and I was told I would not get any heat, but I got better results, and the boilers lasted longer. The increase in the distance from the fire to the shell was a great advantage, and, of course, incidentally I increased the efficiency of the boiler.

David Moffat Myers. In Table 4 of my paper on Tan Bark as a Boiler Fuel, during the efficiency test the temperature inside the furnace was 1100, the temperature in the combustion chamber, under the boiler, was 1475, the flue temperature was 493, and the thermal efficiency was 71.1 per cent.

2 These figures prove that under conditions of good efficiency it is quite possible to have a higher temperature at some distance from the fuel than close to it. The combustion of the gases is simply retarded to a later point of their travel. This might be caused by the combination of a high velocity of draft with a moderate air supply, so that the oxygen does not come into sufficiently intimate contact with the fuel gases in the primary combustion chamber, that is, in the furnace proper. In the case quoted, the CO_2 ran almost uniformly at about 12 per cent, the O between 6 and 7 per cent, with no determinable CO.

A. Bement. In the boiler which Mr. Dean describes, I like the scheme of having the rear end of the chain grate exposed so that it is accessible. The capacities obtained with these boilers are very large; the strength of draft, however, is somewhat too much for an ordinary chain-grate fire. It is my experience that chain grates are not proportioned so that it is possible to carry the requisite thickness of fire for a draft such as existed in this case. I think this will account for the low percentage of CO_2 in the combustible gases, and in this is found the reason why the efficiency was not higher.

2 I would attribute the leaking of the tube ends in the head over the fire to another cause than that given. Considerable experience in similar cases leads me to believe that the trouble is caused by excessive heating on the delicate tube ends in the flue sheet. There are two thicknesses of metal to be penetrated before the heat reaches the water; also the opportunity for water to enter among the tubes and to flow freely over the heated parts is rather restricted. When the ordinary return tubular boiler is set with a fire under the shell, a large portion of the heat flows through the shell, with the result that the temperature of the gases is much reduced, so that by the time they impinge upon the tube sheet, their temperature is low enough so that no damage results.

3 A case of trouble of this kind is illustrated by Fig. 1 and Fig. 2, the first showing a return tubular boiler set against an enclosed fire-brick furnace, in which the gases first impinged upon the tube sheet, passing through the tubes to the other end of the boiler, thence find-

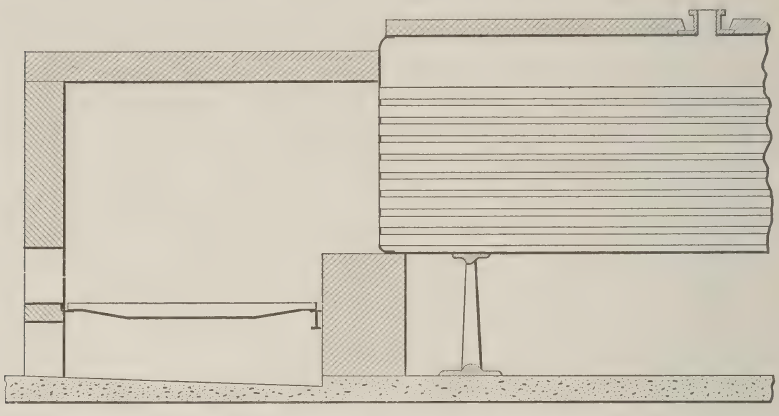


FIG. 1 SETTING OF A FIRE-TUBE BOILER IN WHICH THE TUBES LEAKE

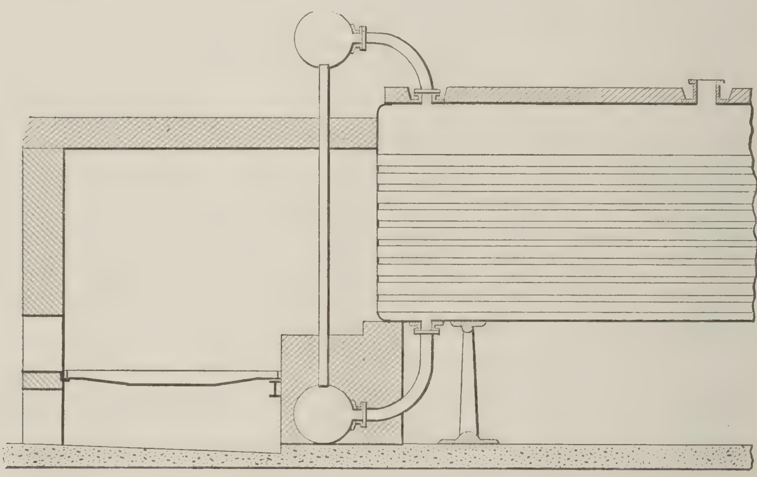


FIG. 2 SHOWING WATER LEG TO LOWER TEMPERATURE OF GASES IMPINGING ON TUBE SHEET

ing exit by way of a chimney attached thereto. When these boilers were put at work immediate and very serious trouble resulted with the tube ends; they leaked very badly, the bead getting out of shape and springing away from the sheet. By means of a little door in the

side of the furnace one could see the water squirting from every tube, and running away from the setting on the floor in a large-sized stream.

4 A remedy was effected in this case, as shown by Fig. 2, by mounting above and below the furnace a drum which extended crosswise of the setting, and connected by vertical 4-in. boiler tubes as indicated; each of these drums being in communication with the boiler, allowed circulation of water and steam. With this scheme the gases first pass between these vertical tubes, which are set closely together, with the result that there is a considerable reduction in the temperature of the gases before they came in contact with the end of the boiler tubes.

5 Another case of this character was remedied by carefully cleaning off the end of the boiler and coating it with an asbestos cement, which was rounded over and into the boiler tube openings in such a way that the flue sheet was entirely protected. This covering lasted about three months, after which it was necessary to renew it. As it was a house-heating boiler, two renewals a season served until the boiler plant was dismantled. The cure of the trouble with the boiler having the extended water leg, as shown in Fig. 2, is due in my opinion to the added heat-absorbing surface in the deeper leg, as it operates to abstract a much larger quantity of heat from the gases before they came in contact with the tube ends, than did the boiler before alteration.

The Author. Replying to Mr. Bolton's remarks, I have heard of the experiments which he quotes in regard to the rate of evaporation of different portions of the length of a tube, but I am not at all impressed with them as a guide. It is well known that the first surface that receives heat gives the greatest rate of evaporation and leaves less for the remaining surface to do. Attention to this to the extent apparently advocated by Mr. Bolton would lead to an absurd result, for one might go on indefinitely shortening tubes. It should be remembered that only 16 ft. of the 20 ft. length of tubes are in contact with water. the remainder being for superheating.

2 Apparently Mr. Bolton believes that it is known how long tubes should be. I do not think that this is known, for the reason that a boiler must undergo a wide range of duty: a short tube would do for light work and a long one would be necessary for heavy work. Many vertical boilers with $2\frac{1}{2}$ in. tubes 20 ft. long have been used successfully for years and they are still being built. Mr. Bolton would evidently prohibit increasing the size of a boiler by increasing the

length of tubes, and would recognize only an increase in diameter as a means of increasing size. To my mind this is illogical and not consistent with the teaching of successful practice.

3 Mr. Bolton speaks of the small combustion chamber as the boiler was first installed, but he ignores the hundreds, if not thousands, of vertical boilers with less combustion chamber space. I believe that I am the only person who designs vertical boilers with the crown sheet as much as 8 ft. above the grate, and this I have been doing for

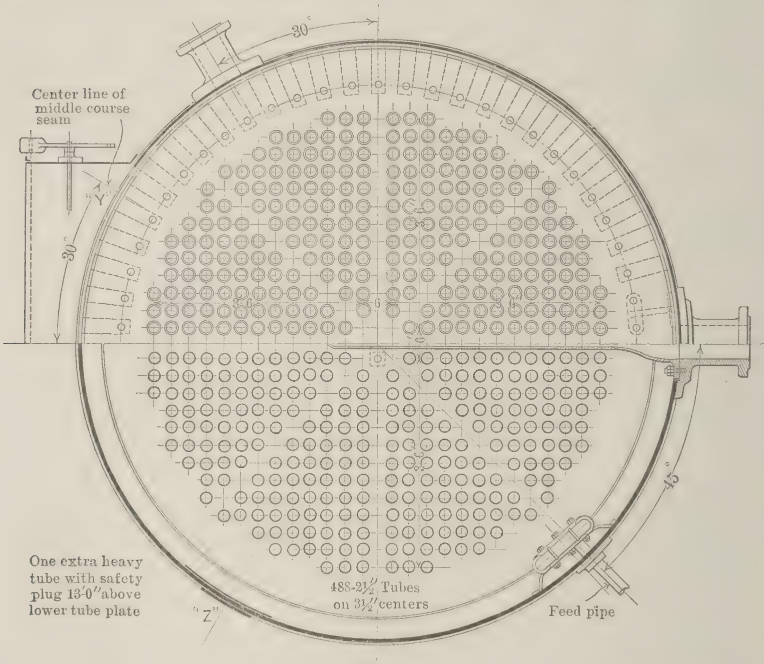


Fig. 1 Cross Section of Vertical Fire-Tube Boiler Designed by the Author

many years. In regard to the Dutch oven in front of these boilers, it would have wholly defeated the object of using vertical boilers. Besides it would have added undesirable brick work.

4 Mr. Bolton easily accounts for the lack of economy of the boiler, but ignores the perfection with which it absorbs heat. I believe the lack of economy to be wholly due to want of air, and when this is supplied and properly distributed the economy will be satisfactory. This would be equally true if the combustion chamber were much longer. The locomotive boilers tested at the St. Louis Exposition by the Pennsylvania Railroad have very little combustion chamber

space, and the excellent economy is due to the proper admission and distribution of air. In regard to the economy of the boilers under

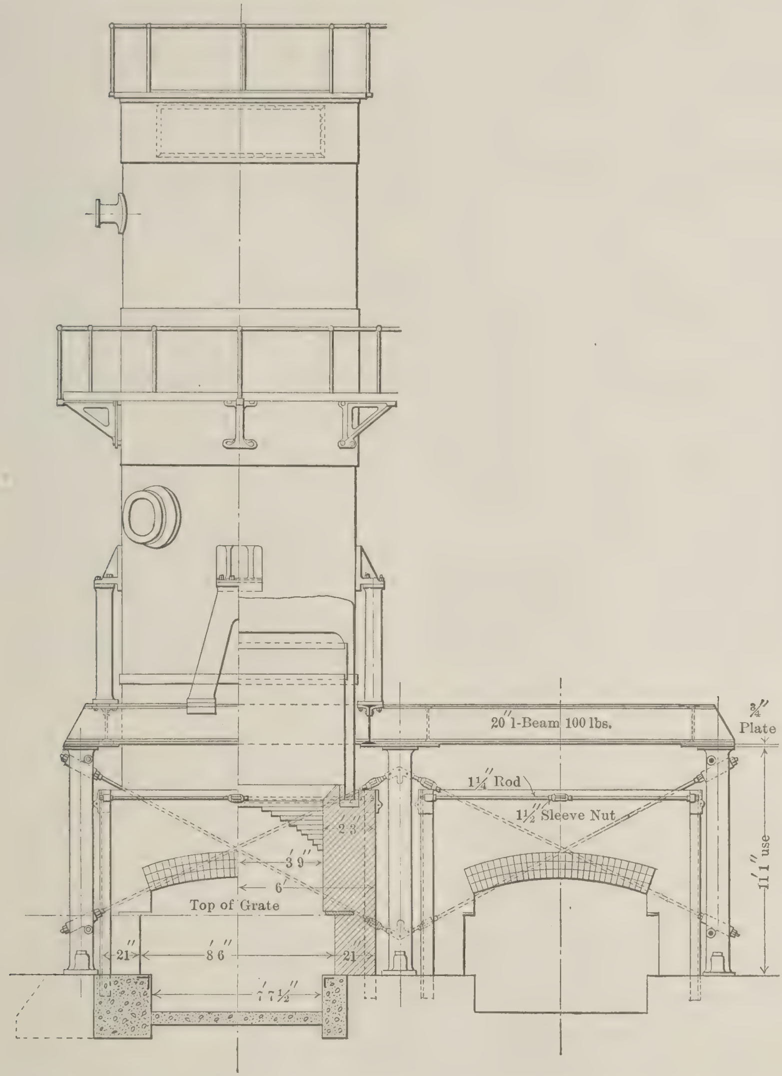


FIG. 2 SECTIONAL ELEVATION OF FURNACE OF THE AUTHOR'S FIRE-TUBE BOILER

discussion, it should be remembered that it was good, only not as good as is sometimes the case.

5 Mr. Parker states that the tubes leaked for the reason that they were set close to the grate and were therefore subjected to wide ranges

of temperature. This is true if we consider the closing of many of the tubes by clinker and the consequent overheating of those that were not closed.

6 I agree with Mr. Bement that some other kinds of stoker would probably not have precipitated the clinker on the tube ends, and this I stated in the paper.

7 Concerning the ability of the water to enter among the tubes, there are many large vertical boilers, some nearly as large as the one described, that have far less space for the passage of water among the tubes, and no trouble results. I know of some that have only one wide space across the crown sheet, while mine have eight wide spaces entirely across, or sixteen reaching to the center.

8 I observe that Mr. Bement considers that the cause of the cessation of the leakage of the tubes of my boiler was the added surface of the water leg. I cannot feel that this is so. It is inconceivable to me that the heat near the center of the furnace should be sensibly reduced thereby. Moreover the absence of the clinker after the change seems to me ample cause of the improvement, for, as I have stated in the paper, a large proportion of the tubes were stopped up, and those that were in service must have been overheated. I think that if the boilers had been raised without adding to the water leg the trouble would have ceased.

9 Whatever the cause of the leakage may have been, I find on

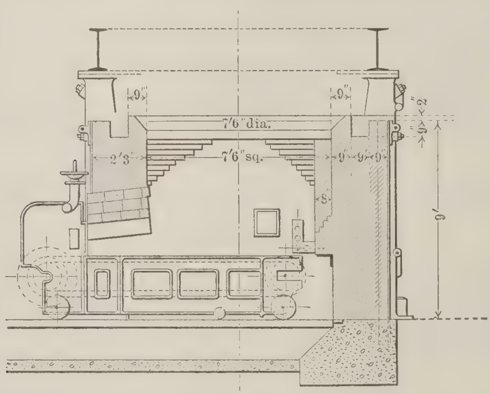


Fig 3 Section of Furnace of the Boiler Shown on page 279

January 17, 1910, the date of writing, that the tubes are not leaking, nor have they leaked since August 31, 1908, in the case of one boiler, and February 25, 1909, in the other, each boiler having been worked constantly to about 1000 boiler horsepower.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M. E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

AERODONETICS. By F. W. Lanchester. *New York, Van Nostrand Co., 1909.*

AMERICAN BRASS FOUNDERS ASSOCIATION. Proceedings of the 3d Convention. *Cincinnati, O., 1909.* Gift of the Association.

AMERICAN FOUNDRYMEN'S ASSOCIATION. Proceedings of 13th Annual Convention. *Cincinnati, O., 1909.* Gift of the Association.

AMERICAN SOCIETY OF AGRICULTURAL ENGINEERS. *Transactions.* Vol. 1. No. 1. December 1907. *Madison, Wis., 1907.* Gift of the society.

CAMBRIDGE BRIDGE COMMISSION. Report. 1909. *Boston, Mass., 1909.* Gift of the Commission.

CARNEGIE TECHNICAL SCHOOLS. General Catalogue. 1909. *Pittsburgh, 1909.*

CIENCIAS MEDICAS E HIGIENE. Vol. 1. (Vol. 1 of the report of the Pan-American Scientific Congress.) *Santiago, Chile, 1909.*

COLUMBIA UNIVERSITY. Annual Report. 1909. *New York, 1909.*

CONCRETE-STEEL CONSTRUCTION. (Der Eisenbetonbau.) *By Emil Mörsch. New York, Engineering News Pub. Co., 1909.*

ELEMENTS OF MACHINE DESIGN. By D. S. Kimball and J. H. Barr. *New York, J. Wiley & Sons, 1909.*

LOCOMOTIVE PERFORMANCE UNDER SATURATED AND SUPERHEATED STEAM. (American Railway Master Mechanics' Association.) 1909.

MODERN GAS ENGINE AND THE GAS PRODUCER. By A. M. Levin. *New York, J. Wiley & Sons, 1910.*

NATIONAL COMMERCIAL GAS ASSOCIATION. Proceedings, 3d and 4th Annual Meetings. *Chicago, 1908.* Gift of the Association.

NOTES ON METHODS AND PRACTICE IN THE GERMAN ELECTRICAL INDUSTRY. (Institution of Electrical Engineers.) By L. J. Lepine and A. R. Stelling. Gift of C. W. Rice.

OSTERREICHISCHEN INGENIEUR- UND ARCHITEKTEN VEREINES. Index to Membership, Vol. 38. *Vienna, 1909.*

PUBLIC WATER SUPPLIES. Edition 2. By F. E. Turneaure and H. L.Russell. *New York, J. Wiley & Sons, 1909.*

REPORT OF THE COMMITTEE ON SMOKE PREVENTION. Presented to the Cleveland Chamber of Commerce, Nov. 16, 1909. Gift of E. P. Roberts.

SOUTH IN THE BUILDING OF THE NATION. History of the Southern States. Designed to record the South's part in the Making of the American Nation. Vol. 1. *Richmond, Va., 1909.* Gift of Southern Publication Society.

SOUTHERN ENGINEER. Vol. 12. No. 4–date. 1909–date.

STRUCTURAL DETAILS OR ELEMENTS OF DESIGN IN HEAVY FRAMING. By H. S. Jacoby. *New York, J. Wiley & Sons, 1909.*

TECHNICAL PRESS INDEX, JANUARY 1908–JUNE 1909. *New York, Technical Literature Co., 1909.*

VIEWS AND DESCRIPTION OF FILTRATION SYSTEM OF THE DENVER UNION WATER Co.

VIEWS AND DESCRIPTION OF LAKE CHEESMAN. The Bulwark of Denver's Water Supply.

WELDING AND CUTTING METALS BY AID OF GASES OR ELECTRICITY. By L. A. GROTH. *New Yokr, Van Nostrand Co., 1909.*

EXCHANGES

AMERICAN SOCIETY OF CIVIL ENGINEERS. *Transactions.* Vol. 65. *New York, 1909.*

CARNEGIE SCHOLARSHIP MEMOIRS. Vol. 1. *London, 1909.*

FUEL TESTS WITH HOUSE-HEATING BOILERS. University of Illinois, Engineering Experiment Station. Bulletin No. 31. By J. M. Snodgrass. *Urbana, Ill., 1909.*

INCORPORATED INSTITUTION OF AUTOMOBILE ENGINEERS. *Proceedings.* Vol. 3. *London, 1909.*

U. S. STEAM ENGINEERING, BUREAU OF. Annual Report. 1909. *Washington, 1909.*

TRADE CATALOGUES

AMERICAN RADIATOR COMPANY, *Chicago, Ill.* Stock list of boilers, radiators, etc., 283 pp.; The Houses Successful, 40 pp.; Ideal Heating, 46 pp.; Results Successful, 20 pp.; Ideal boilers, 32 pp.; Vento cast-iron, hot-blast heaters, 48 pp.; Ideal round boilers, 32 pp.; Ideal sectional boilers, 32 pp.

NEWMAN CLOCK COMAPNY, *Chicago, Ill.* Testimonials concerning the watchmen's portable clok, 9 pp.

NILES-BEMENT-POND COMPANY, *New York, N. Y.* Pond rigid turret lathe, 44 pp.

Sanders, Rehders & Company, Ltd., *London, England.* Sarco patent gas volume recorder, 4 pp.; Recording and indicating steam meters, 4 pp.; Fuel calorimeters, 4 pp.; Patent reversion recorder, 2 pp.; Recording draught gauge, 4 pp.

Underfeed Stoker Company of America, *Chicago, Ill.* Publicity Magazine, December 1909, concerning the Jones Stoker, 16 pp.

Wisconsin Engine Company, *Corliss, Wis.* Bulletin C-4. Heavy Duty Corliss Engines, belted type, 24 pp.

UNITED ENGINEERING SOCIETY

Index to Economic Material in Documents of Maine, 1820-1904; Massachusetts, New Hampshire, New York, Rhode Island, Vermont, 1789-1904. By A. R. Hasse. *Washington, 1907-1908.*

Manufacturers' Record. Annual Blue-book of Southern Progress, 1909. *Baltimore, 1909.* Gift of Manufacturers' Record.

Illustrations of Concrete Steel Arch Bridges. Gift of Concrete Steel Engineering Company.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

08 A company in the Middle West manufacturing moderately heavy machinery has a vacancy for an expert mechanical designer. The position will require a man of considerable experience in the designing of power machinery, familiar with steam and compressed air, and the application of electric motors. Must be capable of submitting original designs to meet conditions. Give experience fully, salary expected, and quote references. Location, Middle West.

09 Mechanical engineer, familiar with the designing and building of steam shovels. Location, Toronto, Canada.

MEN AVAILABLE

14 Technical graduate, experienced in several varied lines of industry, holding executive positions of responsibility during the last eight or nine years, desires to become associated in position of trust with good manufacturing concern, preferably located in the East or Middle West. Best of references.

15 Member, graduate Stevens Institute, eighteen years experience in design and construction of power plants and special machinery; competent to prepare plans, specifications and estimates; desires position with first-class firm of consulting engineers or manufacturing concern.

16 Graduate electrical and mechanical engineer, thirteen years practical experience in testing, inspection and construction work; past five years in charge electrical department and power plant of industrial establishment; executive ability and excellent references; desires to change to a broader field. Qualified for assistant to consulting engineer, assistant superintendent of manufacturing plant or assistant manager. Salary $2500.

17 Stevens graduate '97, extensive shop and drawing room experience, including steel, reinforced concrete, power house and conveying installations. New York city preferred.

18 Associate, experienced in design and installation of mechanical equipment of power stations, railway and industrial buildings, desires position in Middle West; competent executive in field or drafting room. Salary $200 per month, or on percentage basis with consulting engineers.

19 General manager, or assistant, graduate M. E., at present holding similar position, would like to make change; ten years practical experience; good executive ability, best of references.

20 Member, thoroughly experienced in the design of large gas engines for all services; desires position as chief engineer with company building this class of machinery or with a company desiring to enter the field, preferably the latter.

21 Junior, age twenty-nine, technical graduate, seven years experience shop, foundry, drawing room and office: desires to make a change. Compressed air, pumping machinery, or similar line preferred.

22 Junior member, technical graduate, nine years experience in drafting, construction and office work with engineers and contractors, in engineering departments of industrial companies; wishes to make change after May first. Prefer position with firm of engineers and contractors or with industrial company.

23 Student member, graduating Cornell University June 1910, desires position with engineering concern in New York city or thereabouts. Salary no object. Can furnish highest references.

24 Graduate of M.I.T., in electrical and mechanical engineering; experience in design and construction of machinery and buildings; development of systems and organization; making of reports, compilation of data, etc.

COMING MEETINGS

February-March

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the Editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN ASSOCIATION OF RAILROAD SUPERINTENDENTS
March 18, Chicago. Secy., O. G. Fetter.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
February 11, monthly meeting, 29 W. 39th St., New York. Paper: A Modern Automatic Telephone Apparatus, W. Lee Campbell. Secy., R. W. Pope.

AMERICAN INSTITUTE OF MINING ENGINEERS
March 1-5, Spring meeting, Hotel Shanley, Pittsburg, Pa. Secy., R. W. Raymond, 29 W. 39th St., New York.

AMERICAN MATHEMATICAL SOCIETY
February 26, New York and San Francisco sections. Secy., F. N. Cole, 501 W. 116th St., New York.

AMERICAN RAILWAY ENGINEERING ASSOCIATION
March 14-17, Chicago. Secy., E. H. Field, Monadnock Bldg.

AMERICAN SOCIETY OF CIVIL ENGINEERS
February 2, 16, 220 W. 57th St., New York, 8:30 p.m. Papers: Underpinning the Cambridge Building, New York City, by T. K. Thomson; Effect of Alkali on Concrete, by G. G. Anderson. Secy., C. W. Hunt.

AMERICAN SOCIETY OF ENGINEERING CONTRACTORS
February 24-26, annual meeting, Chicago, Ill. Secy., Daniel J. Haner, Park Row Bldg., New York.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS
February 8, 29 W. 39th St., New York. February 16, City Club, Boston, May 31-June 3, Spring Meeting, Altantic City, N. J. July 26-29, joint meeting with Institution of Mechanical Engineers, Birmingham, England. Secy., Calvin W. Rice, 29 W. 39th St.

ASSOCIATION OF ONTARIO LAND SURVEYORS
February 22-24, annual meeting. Secy., Killaly Gamble, 703 Temple Bldg., Toronto.

BOSTON SOCIETY OF ARCHITECTS
February 1, Parker House, Boston, Mass. Dinner in the Crystal Room at 6.30 p. m. Secy., Edwin J. Lewis, Jr.

CANADIAN FORESTRY ASSOCIATION
March 10-11, Fredericton, N. B. Secy., Jas. Lawler, 11 Queen's Park, Toronto, Ont.

CANADIAN MINING INSTITUTE

March 2-4, annual meeting, Toronto, Ont. Secy., H. Mortimer-Lamb, Windsor Hotel, Montreal.

CONNECTICUT SOCIETY OF CIVIL ENGINEERS

February 8, annual meeting, New Haven, Conn. Secy., J. Frederick Jackson, Box 1304, New Haven, Conn.

ENGINEERING SOCIETY OF WISCONSIN

February 23-25, Milwaukee, Wis. Secy., W. G. Kirchoffer, 31 Vroman Bldg., Madison.

ENGINEERS CLUB OF PHILADELPHIA

February 5, annual meeting, 1317 Spruce St. Secy., W. P. Taylor.

INSTITUTION OF MECHANICAL ENGINEERS

February 18, Institution House, Storey's Gate, St. James' Park, Westminster, S. W., London, England. Secy., Edgar Worthington.

IOWA ASSOCIATION CEMENT USERS

March 9-11, Cedar Rapids. Secy., Ira Williams, Ames.

IOWA ENGINEERING SOCIETY

February 16-17, Cedar Rapids. Secy., A. H. Ford, Iowa City.

MINNESOTA ELECTRIC ASSOCIATION

March, St. Paul. Secy., B. W. Cowperthwait.

NATIONAL ASSOCIATION OF CEMENT USERS

February 21-26, annual convention, Chicago, Ill. Pres., Richard L. Humphrey, Harrison Bldg., Philadelphia, Pa.

NEBRASKA CEMENT USERS ASSOCIATION

February 1-4, Lincoln. Secy., Peter Palmer, Oakland.

NEW ENGLAND ASSOCIATION OF GAS ENGINEERS

February 16, 17, annual meeting, Boston, Mass. Secy., N. W. Gifford, 26 Central Sq., East Boston, Mass.

NEW ENGLAND RAILROAD CLUB

March 8, annual meeting, Boston, Mass. Secy., George H. Frazier, 10 Oliver St.

NEW ENGLAND STREET RAILWAY CLUB

March 24, annual meeting, Boston, Mass. Secy., J. J. Lane, 12 Pearl St.

NEW ENGLAND WATERWORKS ASSOCIATION

February 9, Hotel Brunswick, Copley Sq., Boston. Papers: Depreciation, L. G. Powers; The Purchase of Coal on Efficiency Basis, A. O. Doane.

NORTHWESTERN CEMENT PRODUCTS ASSOCIATION

February 18-21, annual meeting, Chicago, Ill. Chairman notification committee, O. U. Miracle, Minneapolis, Minn.

PACIFIC COAST ELECTRIC AUTOMOBILE ASSOCIATION

February, Oakland, Cal. Secy., J. T. Halloran, 604 Mission St., San Francisco.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

February 15, West Hall of the Rhode Island School of Design, 8 p. m. Paper: The Bliss-Levitt Torpedo, Samuel Aronson and Chas. Gabriel.

RAILWAY SIGNAL ASSOCIATION

March 14, Chicago. Secy., C. C. Rosenberg, Bethlehem, Pa.

SCRANTON ENGINEERS CLUB

February 3, annual dinner, Club Rooms. Secy., A. B. Dunning.

SOUTHERN GAS ASSOCIATION
February 16, Chattanooga, Tenn. Secy., James Ferrier, Rome, Ga.

STEVENS ENGINEERING SOCIETY
February 8, 15, 21, Hoboken, N. J. Papers: Pavements for City Streets, Samuel Whinery, Mem. Am. Soc. M. E.; Power Plant Economies, D. S. Jacobus, Mem. Am. Soc. M. E.; The Engineer as a Manager, H. L. Gantt, Mem. Am. Soc. M. E. Secy., R. H. Upson.

UNIVERSITY OF CINCINNATI, Student Branch, AM. SOC. M. E.
February 18, regular meeting. Paper: Milling Machines and their Uses, C. S. Gingrich, Jun. Am. Soc. M. E. Secy., P. G. Haines.

MEETING IN THE ENGINEERING SOCIETIES BUILDING

Meetings of All Kinds

Date	Society	Secretary	Time
February			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engrs. and Archs	Z. de Németh	8.30
8	The American Society of Mechanical Engrs	Calvin W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15
March			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engrs. and Archs	Z. de Németh	8.30
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10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15

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Terms expire at Annual Meeting of 1911

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1909

On a Standard Tonnage Basis for Refrigeration

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A. P. Trautwein
Philip De C. Ball
E. F. Miller

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John E. Sweet
H. H. Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

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JESSE M. SMITH (3)

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ALEX. C. HUMPHREYS FRED J. MILLER

NOTE—Numbers in parentheses indicate number of years the member is yet to serve.

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James D. Andrew
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STUDENT BRANCH	AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	....	C. F. Hirshfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	C. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	E. A. Rogers	H. C. Warren
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. A. Kirk	J. R. Jackson
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	H. S. Coleman	John Garver
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	W. F. Colman	S. G. Wood
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	....	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Fredk. A. Dewey	A. P. Truette
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	W. H. Mantgomery	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	R. N. Trane	G. A. Glick
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. E. Dudley	F. T. Kennedy
Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	....	....

DEDICATION OF MEMORIAL TABLET TO ROBERT HENRY THURSTON

A bronze memorial tablet to Dr. Robert Henry Thurston, first president of The American Society of Mechanical Engineers, was dedicated at the New York monthly meeting, Tuesday evening, February 8, 1910, in the auditorium of the Engineering Societies Building, in the presence of many associates and former students of Dr. Thurston as well as of members of the Society. This bas-relief, which is the work of Herman A. MacNeil, a former student and personal friend of Dr. Thurston, and is a replica of the memorial tablet presented to Sibley College, Cornell University, by alumni and students, was placed in the rooms of the Society through the generosity of members, as an expression of their devotion to Dr. Thurston. The contributions were received by a committee consisting of John Fritz, S. W. Baldwin, Prof. R. C. Carpenter, Walter C. Kerr, E. A. Uehling, Wm. Hewitt and Gus. C. Henning; and the details connected with the acquiring of the tablet, its installation and the arrangement of the dedicatory exercises, were in the hands of Dr. Alex. C. Humphreys, *Chairman*, Chas. Wallace Hunt, Fred J. Miller, Prof. R. C. Carpenter and J. W. Lieb, Jr.

The program of the evening was designed to cover the various phases of Dr. Thurston's brilliant career, treated in each case by a speaker of wide reputation who had known Dr. Thurston intimately during this period of his life. It therefore very appropriately included an address on Dr. Thurston's relationship with the Society, by Prof. John E. Sweet, President of the Society from 1883–1884 and active with Dr. Thurston in its organization; a communication on Dr. Thurston's career as a naval engineer from Rear-Admiral Benjamin Franklin Isherwood, U.S.N., Retired, Honorary Member of the Society, which was read by Prof. F. R. Hutton, Honorary Secretary; an address on Dr. Thurston at the Naval Academy at Annapolis, by Rear-Admiral George W. Melville, U.S.N., Retired, Honorary Member and Past-President, Am.Soc.M.E.; on Dr. Thurston as professor at Stevens Institute of Technology, by Col. E. A. Stevens,

trustee and treasurer of Stevens Institute and son of its founder; on Dr. Thurston's literary and research work, by William Kent, one of the organizers of the Society and a close friend and co-worker with Dr. Thurston; and on Dr. Thurston as director of Sibley College, Cornell University, by Walter C. Kerr, a trustee of Cornell.

After the addresses of the evening, members and guests proceeded to the eleventh floor where the tablet was unveiled and presented by Dr. Humphreys, on behalf of the committee, to the Society, for whom it was accepted by Col. E. D. Meier, Vice-President. Col. Meier cites this as the first bronze statue of an eminent engineer to be erected in the United States in a great building devoted entirely to engineering and said that an excellent choice had been made of Dr. Thurston as a representative of his profession. Dr. Humphreys also presented Herman A. MacNeil, the artist, to the audience, who made the concluding remarks of the evening.

The addresses of the evening follow.

It was a matter of regret that Mrs. Thurston found it impossible to be present at the meeting. A letter was read by the Chairman, expressing her appreciation of the honor rendered to Dr. Thurston. Messages were also received from President Westinghouse, who was prevented by urgent business from attending, and from Chief Engineer Chas. H. Manning, U. S. N., and Lieut-Commander Robert Crawford, U. S. N., associated with Dr. Thurston in the Naval Academy at Annapolis.

The Chairman of the Thurston Memorial Committee, Dr. Alex. C. Humphreys, president of Stevens Institute of Technology, presided over the meeting, and said in his introductory remarks:

REMARKS BY DR. ALEX. C. HUMPHREYS, CHAIRMAN

While recognizing that it is not the function of a presiding officer to forestall the speakers to be introduced, I cannot refrain from saying a few words about my friend and preceptor, Robert H. Thurston. Others will tell you of his widely varied activities, his tremendous capacity for work, which was nevertheless overtaxed, his quickness of brain and speech, his powers of exact determination and expression, his capacity for organization and execution, his eminence as an engineer and educator. I prefer to think of him as the large-hearted, gentle, lovable, helpful man; the man of vision, the optimist.

While a student at Stevens, I was not fortunate enough to have Dr. Thurston's guidance during my junior year, for then, in 1879, he had

not yet recovered from the almost fatal nervous breakdown, which resulted from his strenuous life in many lines of activity. But I came to know him well during my senior year, and had many occasions to be deeply grateful to him for his assistance and encouragement, which I then greatly needed. I never saw him other than cheerfully responsive to a request for help, and I was never allowed to feel that I was intruding when I went to him for counsel. While demanding respect and obedience from those under him, his attitude towards them was characterized by a sympathetic desire to be helpful.

Wm. Kent, one of the speakers of the evening, in his masterly biographical notice of Dr. Thurston in the Sibley College Journal, in writing of the vast amount of work performed at a certain time by Dr. Thurston, says: "And during all this time, I never saw him excited or ruffled over his work." We busy, overcrowded men, know this to be high praise indeed.

I met Thurston too seldom after I graduated from Stevens, but when we did meet, I was made to feel that he was really interested in my career, and that he rejoiced and sympathized with me as circumstances suggested. I like to remember that he came down from Cornell at the time of my inauguration as president of Stevens Institute, and that it was through him that Cornell University and Sibley College conveyed their good wishes to Stevens Institute and to me at that time. Later in that year he quietly passed away to his well-earned rest.

Thurston was a man of vision. Time and again this is shown in his writings, and especially in view of later developments. And this, notwithstanding that his declared results were sometimes afterwards to be amended, as must be the case of those who are courageous enough to act the part of pioneers.

We are, apparently, now only beginning to appreciate in this country the practical and commercial value, to say nothing of higher things, of technical and technological education. And even now, those who do have the appreciation are unable to move and guide those who have the power to provide the means for the necessary improvement in our educational methods. Years ago, Thurston wrote: "Germany has substituted for the now obsolete apprenticeship system, the systematic, scientific methods of preparing her youth for the future of their lives in all departments of instruction and industry."

He was a student of political economy and education and pointed out the evils which would come to us unless certain lines of reform were followed, the evils which are now upon us and have to be met by patience, wisdom, firmness and common sense.

It was said of his father: "Throughout his life, his benevolence, his uniform kindness to employes and to all with whom he came in contact, and his strong attachment to his friends, made him as universally beloved as he was widely known."

The son was strong in faith though he did not carry his religion in his sleeve. He gave voice to his faith in a certain article which he wrote under the title. The Scientific Basis of Belief, which received wide attention. It seems to me that the summation of his creed is found in a verse which he included in this article:

Strong Son of God, immortal love,
  Whom we, that have not seen Thy face,
By faith, and faith alone, embrace,
  Believing where we cannot prove.

Notwithstanding his great and varied accomplishments, it is as the holder of this faith, and as the worthy son of this worthy father, that I love to think of Robert H. Thurston.

DR. THURSTON'S CONNECTION WITH THE SOCIETY

By John E. Sweet

Honorary Member and Past-President, Am.Soc.M.E.

We meet tonight to do honor to the first president of The American Society of Mechanical Engineers, elected now nearly thirty years ago. I have been asked to tell the simple story of his connection with the Society. It is fitting that we whose fading memories can give only shadowy reviews of past events do the best we can to record the facts as we recall them.

To begin at the beginning we must hark back to the fall of 1879, when the American Machinist was published in a small office at 96 Fulton St., New York. The journal had been in existence but a few years. It had received contributions from a goodly number of contributors engaged in various branches of mechanical industries and from a wide section of the country. But very few of these contributors were known to the publishers, and fewer still to one another, and the notion came to my mind to get as many of them together as we well could, and give the publishers a surprise party; with a faint notion that it might lead to an organization. I conveyed the notion to one of the contributors, and he at once gave it away to the editor, with the suggestion that some sort of a mechanical association be formed. The suggestion took root in the minds of

the publishers, and Mr. Bailey, the editor, came to Syracuse to see me about it; or, in fact, to inveigle me into writing the invitations.

Among those invited were Alexander L. Holley and Prof. Robt. H. Thurston, then in a sanitarium in Dansville, N. Y., both of whom entered heartily into the scheme. Before the meeting, which was held on February 16, 1880, Mr. Bailey and I had an interview with Mr. Holley; each was to draw up some form of program for the meeting, and we were to meet the next day to compare notes. As such things usually turn out, Mr. Holley had drawn up a set of rules which were so complete that we could readily endorse them. At the meeting in the afternoon there were something like thirty present, with letters of endorsement from fifteen or twenty others. Mr. Holley acted as chairman, and I well remember the point he made in his opening address, that it had come to that state of affairs that both civil and mining engineering were largely mechanical. A good deal of time was spent in discussion of the rules, which ended in the adoption of those Mr. Holley had prepared; and time was also wasted in settling on a name, until Mr. Copeland said, "Call it 'The Society of Mechanical Engineers." This seemed to settle it, except that in the shuffle the word "American" got incorporated, to the regret of possibly no one but myself.

Mr. Copeland, Charles T. Porter, Mr. Holley, E. D. Leavitt, Jr., and myself were chosen as a nominating committee. The officers nominated were elected at the meeting held April 7, 1880, at the Stevens Institute, over which Henry R. Worthington presided.

At the time of what is now known as the first annual meeting, held in New York, November 4 and 5, 1880, Professor Thurston had regained his health, and was able to preside and to deliver an able address. Professor Thurston was elected president for the second time, and these two were the critical years in the Society's history. We then held three meetings a year, and while Holley and Worthington lived, they formed with Professor Thurston a three-point support that did not rock; but they both died while Thurston was President, and left him to carry the burden. One incident that occurred during this time I shall always remember. Going out from one of the meetings Mr. Worthington, greatly elated over the way things were moving, said to me, "Professor, the thing is going to go." I doubt if any of us had the idea that the Society would reach a membership of 300, while today it takes another right hand cipher to record our membership.

Professor, later Dr. Thurston, not only while president, but for a

long time later, was more instrumental in helping up the Society than any other man. It is not necessary to enumerate his contributions to the Transactions. He never showed evidence of elation at success or chagrin at defeat. His work enriches every volume of the Transactions, from the first volume down to the time of his death. And every member of the Society needs to open the door of his memory and let the history of its work shine in and enliven his spirit of respect and adoration for the boy, the student and the scholar, the thinker and the worker, the teacher and the guide, the honored member and revered first President of The American Society of Mechanical Engineers, and the man—Dr. Robert Henry Thurston.

DR. THURSTON'S CAREER AS A NAVAL ENGINEER

By Rear-Admiral Benjamin F. Isherwood

Engineer-in-Chief, U.S.N., Retired, Hon. Mem.Am.Soc.M.E.

Professor Thurston, in whose honor these commemoration exercises are held, was in all respects an ex eptional person, with endowments not only of a very rare but of a very high order. He was a typical representative of the American engineer of the present day; combining a thorough and extensive practical knowledge of his profession with a scientific culture scarcely found in the exclusively theoretical scientist; and he had, in addition, the ability to make these qualifications available to the world by means of an excellent literary education improved by a carefully discriminating practice as a writer, an orator, a mathematician, and an original investigator in the broad field of his profession. The first-class engineer of the present day must also be a first-class scientist as well as a first-class mechanician, besides possessing a mind well stored with the information collected by others of his profession whose aims and achievements are similar to his own.

With these mental powers was associated, in the case of Professor Thurston, so charming a personality that he not only never had a foe, but all who knew him were his friends. His knowledge and his services were at the command of all who sought them, and were rendered in a manner that made the recipient believe that instead of receiving a favor, he was conferring one. Professor Thurston was the author of several books on engineering subjects, which were classics in their day. He was also a prolific investigator of difficult phenomena in engineering, and his numerous reports have much

enriched its literature and enlightened its obscurities. He wrote with perspicuity, elegance and ease; and he was a ready and fluent orator on all the scientific topics of the day.

His death was a great loss to the world, and particularly to his own profession of engineering, for his exceptionally valuable life was devoted to the improvement of the world in the only way it could, in his opinion, be improved, namely, by the cultivation of physical science. Great intellectual attainment meant with him, great everything else.

Those who knew Professor Thurston best valued him the highest. My personal acquaintance with him was long and intimate, and it was intensified by our professional interest in the same subjects, notwithstanding the great difference of our ages and temperaments; and none who knew him as well as I did will consider this weak portraiture of him as overcolored. His death at Cornell saddened all who knew him well enough to appreciate his gentle qualities, as well as his lofty aspirations. He was most happily constituted; he lived his life in the sunshine of an entirely normal existence, and by dying in the full flush of manhood and in the consciousness of great achievement, he was saved from decline, and enabled to pay to glory the debt he owed to nature.

DR. THURSTON AT THE NAVAL ACADEMY AT ANNAPOLIS

By George W. Melville, Rear-Admiral, U.S.N., Retired
Honorary Member and Past-President, Am.Soc.M.E.

My function is to pay the tribute of the navy to one who was for a time a naval officer and who during his career as such bore himself in a way worthy of its best traditions, and left a record the memory of which is still distinct with those of us who were his contemporaries although it has long since been overshadowed by the greater reputation of his more mature life.

When the engineer came into the navy, he received scant recognition, although we were very fortunate indeed in having as our first representative that grand old man, Charles H. Haswell, who was taken from us so recently. The dominant faction in the navy did not like machinery nor mechanics, and as a result the very early engineers were largely men whose theoretical training did not go beyond the common schools and whose professional training came from hard knocks in the machine shop. This was true in my own

case and in that of the great majority of the older engineers. At the breaking out of the Civil War, however, with the increased demand for engineers and the desire of patriotic men with engineering training to render their best service to the Government, a number of men came into the corps who were college graduates, and Dr. Thurston was one of these. He entered the service in July 1861, very soon after the beginning of hostilities and before any of the great naval battles, and he served at sea continuously until the close of the war, taking part in the Battle of Port Royal and in the Siege of Charleston. While still a second assistant engineer, he was placed in charge of the machinery on the Chippewa and later served on the monitor Dictator, the largest built up to that time.

The historian of the engineer corps of the navy, Past Assistant Engineer (now Captain) Frank M. Bennett has mentioned several instances in which Thurston distinguished himself. One of these was on January 29, 1863, when he was in command of one of the armed boat's crews which captured the blockade runner Princess Royal at Charleston. The next day two armored rebel rams came down upon the Federal fleet and destroyed some of the converted merchantmen which constituted it. Thurston was temporarily chief engineer of the Princess Royal and by extraordinary efforts managed to get the machinery going so that she got out to sea and escaped destruction at the hands of the rebels.

With his fine preliminary training at Brown University and his four years of practical experience in the navy, it was obvious that he was well equipped for duty as an instructor at the Naval Academy in the department of natural and experimental philosophy, to which he was ordered in 1865. During his term of service there, the head of the department died and Thurston was made acting head of the department.

It was during this time that the education of engineers at the Naval Academy began with the class of acting third assistant engineers who entered in 1866, of whom the late Admiral Rae was a member. Thurston undoubtedly gave instruction to all of these young engineers in that department, which we would now call thermo-dynamics, although he was not a member of the department of engineering which looked after the more practical side of their professional training.

A retired engineer officer, who was an instructor at Annapolis in the department of engineering, while Dr. Thurston was in the department of physics, speaks of him as follows: "During about two years

of his term at Annapolis, I saw him almost daily and am therefore able to bear testimony to the excellent work he then did both as an instructor and as the managing head of that very important department of our Naval School. He fully appreciated the importance of physical science to the naval profession generally, and its particular application to naval engineering, and he worked with untiring zeal for its full development as a part of the training of our young officers, often personally designing special apparatus, which were fewer then than now, for demonstrating physical truths and principles. Thurston was eminently fitted for this class of scientific work by taste, education and practical experience as a naval engineer.

At such a time as this, we realize how very interesting it would be if we could know exactly what circumstances lead to the selection of a man for a particular line of work. I have tried hard to find just what led to Dr. Thurston's selection for the chair of mechanical engineering in Stevens Institute, but it occurred so long ago, almost forty years, that the details are no longer available. Indeed it is probable that there never was any record of them and that it was a matter between President Morton and Dr. Thurston himself. Unless a man writes an autobiography, and is as frank about all the occurrences of his life as Herbert Spencer, such an interesting phase as this is apt to be entirely passed over in spite of its great importance. Aware, as those of us who knew him well are, of Thurston's marked ability as a scientific expositor and of his life-long desire for progress and increased efficiency, may we not imagine that, before he was asked to become President of Stevens, President Morton had become acquainted with Dr. Thurston through his articles; and by correspondence or conversation had found that here was a man after his own heart, who could be counted upon to make the new school of technology what he wanted it to be, the best in the country.

There were other departments of colleges and perhaps other schools where mechanical engineering was taught, but Stevens was, I believe, the first institution in our country devoted exclusively to the education of mechanical engineers, and we can now realize even better than when he was called to the Chair, how extremely important was the selection of Dr. Thurston for this work. If he had been content to drift along in accordance with old established methods, or if he had not been a tremendous worker, his great ability would have failed to give to Stevens the foremost standing which it has held from the very start.

Dr. Thurston had, in a degree rarely found among men devoted

to education, a strong touch of the commercial instinct, and it was doubtless this which led him, from the beginning, to direct the work of his students in their experiments along lines of an immediately practical interest to engineers and others. For example, I remember a set of experiments to determine the economy of gas engines at a time when they were just coming into use. This was only one of a great many instances. The hard-headed, practical manufacturers could not fail to realize that a school where the energies were directed in such a practical way must turn out men who would make good in practical life. I need hardly show that this judgment has been verified by recalling the positions now held in offices of the very highest importance in the lines of manufacturing and transportation by many of these graduates.

So great was the reputation made by Dr. Thurston at Stevens, that he was generally looked upon as the greatest teacher of mechanical engineering in the country; and I happen to have information with respect to the circumstances under which he went to Cornell which show that when they were looking for a man of the highest accomplishments to take charge of Sibley College, they at first did not consider Dr. Thurston, for the reason that they did not believe any inducement could take him away from Stevens. When the Trustees of Cornell came to the conclusion that they wanted the best man available, and that they were willing to pay whatever was necessary to secure his services, they eventually opened negotiations with Dr. Thurston, which led to his finally going there.

Of his splendid work at Cornell others will speak, and I need only say that it is a marvelous tribute to his ability and reputation that he should have been able to increase the attendance at Sibley College from about one hundred students to over one thousand.

His educational work was so engrossing that it naturally left him little time for keeping up his association with the Navy although we who remained in the service always felt that in him we had a sincere friend on whom we could depend for such help as was in his power to give whenever the engineers of the Navy needed assistance. Before we finally attained the full recognition of the paramount importance of engineering in the Navy, which, as you know, is now the function of the entire line of the Navy, we had many an up-hill fight, and on several occasions Dr. Thurston materially assisted us by articles in the magazines and by personal appeal. In this connection it is interesting to note that Bennett's History of the Steam Navy, in speaking of an order of the Navy Department in

1870, which was considered very unfair to the engineer corps, mentions that, in consequence of this order, a number of the brightest men in the corps resigned, among them Dr. Thurston. It has been my observation that the participants in an up-hill fight are drawn more closely together than those whose association is always on the winning side. It was doubtless Dr. Thurston's keen recollection of the lack of recognition which he had received while an engineer in the Navy, that made him so willing to help those who were still in the service in their efforts for a proper recognition and consideration of engineering and its exponents.

It has been my desire as a naval engineer and one whose whole life has been spent in the naval service, to voice, in behalf of myself and my colleagues who were and are engineers in the Navy, our admiration for the friend who, during his own short naval career, did so much to add to the reputation of the engineer corps, and by his prominence as an engineer all through his life reflected the highest credit upon naval engineering.

DR. THURSTON AT STEVENS INSTITUTE OF TECHNOLOGY

By Col. E. A. Stevens

Member of the Society, Trustee and Treasurer, Stevens Institute

Thurston's work at Stevens can be divided into two parts: on the one hand his general work as an engineer, including his well known contributions to the literature of engineering and the researches on which they rested, and on the other his share in the development of the work of the Institute and the influence of his personality on his fellow instructors and the undergraduates.

As to the first part I can say but little in the time assigned. The history of mechanical engineering in the United States will always bear witness to his ability, to his untiring energy and to the liberality with which he freely gave to his beloved profession all that his ripe experience and trained observation could give.

Whatever may have been the value of his other work while at Stevens, none of it surpasses, or I may say equals in importance, his share in the development of the system of instruction and the influence of his personality and of his standard of professional ethics on those with whom he was there thrown into contact.

Forty years ago the American mechanical engineer was mainly

the product of the shop and the engine room, with such self-teaching as could be gathered in the leisure hours of a busy life of hard work. Most engineers of that day would admit that draughting and mathematics could be taught in schools, but claimed that such training would produce draughtsmen and mathematicians, not engineers, men who would be of less value in practical work than the lad of the same age who had spent his time in the shop; that the school-bred man would need several years of hard work to knock the school-taught nonsense out of his head, always granting that he had not been irretrievably ruined by his scholastic training.

Such was the general, even if not the unanimous mind of the profession when Henry Morton gathered around himself six men, who with him were to form the faculty of the first American school wholly devoted to the teaching of mechanical engineering. Scientist and scholar as he was, Morton appreciated the gravity and importance of the task set him and selected his fellow members of the faculty with a care and judgment amply justified in the result. Of these men, eminent as they were, Thurston was the one on whom devolved the practical teaching of engineering. The others must have aided, and unquestionably did aid, in giving the training as a whole a practical direction, but it was to Thurston more than to any one other of Morton's first faculty that the prominence of the practical curriculum at Stevens must have been due, and on him therefore it is but fair to bestow a generous share of the acknowledgment due these men. It would be as invidious as it would be useless to apportion to each the share due to his individual efforts.

While Thurston's personality impressed itself on all who met him, whether at Stevens or elsewhere, the lasting result of this impression on the men who there studied under and with him forms a part of the history of Stevens. The material that came to the "Old Stone Mill" was much the same in the early days as since. The early graduates at once took a standing in American engineering work that soon settled once and for all any debate as to the value of a technical training. They carried also with them into the world what was as necessary for the progress of engineering as technical skill or practical knowledge. They had imbibed together with their calculus and thermodynamics that moral and ethical view of their profession without which an engineer's skill and learning is of little value to his country, a thing not absorbed from text books or taught by platitudes, however often reiterated from the lecture platform.

Boys and young men quick to detect cant are equally quick to

recognize and value square-dealing and to love and follow and model themselves after the straightforward man. Of all of that first faculty there was no one to whom the undergraduates could and did more confidently look for a square deal than to Thurston. That straight clear gaze, right into your eye, gave at once a confidence in the man and in his methods in and in a feeling of sympathy that experience did not belie.

Single examples prove few cases and a life such as Thurston's is not to be judged by citing examples and incidents. The true measure of his great work and usefulness is to be fixed by the standard set by the great Master, "by their fruits ye shall know them." By no other standard would Thurston have asked to be judged and the fruits of his work at Stevens are proven not by the accomplishment of specially gifted men who studied under him but by the general standing of the Stevens men of this day.

DR. THURSTON IN LITERATURE AND IN RESEARCH

By Wm. Kent, Mem.Am.Soc.M. E.

My acquaintance with Dr. Thurston began near the end of the year 1874, when I called upon him to make arrangements for entering the junior class in the Stevens Institute of Technology. He was then thirty-five years of age. He was at this time professor of mechanical engineering, meeting his classes two hours a day for, I think, five days in the week; he was editing the four volumes of reports of the United States Commission to the Vienna Exhibition of 1873, one of the volumes being written by himself; he had shortly before written a report of the United States Commission for investigating the causes of steam-boiler explosions; he was planning the researches to be made by the United States Board to test iron, steel and other metals, of which he was secretary and the most active member. Besides all this he was writing papers for the American Society of Civil Engineers and for the Journal of the Franklin Institute, concerning the results of his researches. In 1871 he had conducted a series of boiler tests on several different makes of water-tube boilers at the American Institute fair in New York. In 1873 he had organized a mechanical engineering laboratory for the purpose of making engineering researches, the first of the kind to be established in the United States. At about the same time he invented his well-known autographic testing machine for testing materials by torsion, and

later he invented the machine for testing lubricants, in which some of the principles of the torsion machine were embodied.

In June 1875, Dr. Thurston called me into his office and told me he wanted me to undertake a research into the strength and other properties of the alloys of copper. I said to him, "I don't know anything about alloys." "That is a good qualification," said he, "you won't have anything to unlearn." He told me how to make a research into the literature of the subject, and how to find indexes to such literature. He had me write him a report of all I could find that was then known about the alloys of copper and tin and copper and zinc, and after studying it he planned a series of tests to be made in the laboratory, which took eighteen months to complete. During all this time I had to report to him almost every day, and I had a desk in his office. Then began an intimate friendship which lasted until the day of his death. During these two years of companionship I was ever more and more impressed with Dr. Thurston's genius and with the breadth of his intellectual power. Not only was he a tireless worker, driving the pen or pounding the typewriter hour after hour, but his brain always seemed to be working as steadily and as rapidly as his pen. Whenever he was asked a difficult question the answer seemed to come instantly from his well-stored mind, and the answer was right. Such a combination of industry, rapid and clear brain-action, and broad intellectual grasp of a great variety of subjects, engineering and other, I have never known in any other man.

Such intense mental activity as Dr. Thurston exhibited in these years led to its natural result, nervous exhaustion. There was a time in 1876 when he visited the Institute only for a few minutes each day, and five minutes conversation on any technical question would almost prostrate him. During this time he was worried by the fear that Congress would not continue the appropriation for the work of the United States Test Board, and he undertook to write a letter on the subject to one of the senators, but it took him a week or more to write the letter working on it five minutes a day. He gave me a copy of it to take to one of the members of the board in New York City, and that member said to me that it was by far the best presentation of the subject that had ever been made. Such was the quality of his work when he was on the verge of physical collapse.

Here is another example of the kind of work he could do during the same period. He had been planning a series of tests of the triple alloys of copper, tin and zinc, and one day on one of his brief

visits to the institute he said to me: "Here is something I want to show you. Here is an equilateral triangle. It is one of the properties of this triangle that if from any point within it perpendiculars are drawn to each of the sides the sum of these perpendiculars is equal to the altitude. Now let us mark one apex 100 copper, another 100 tin, and the third 100 zinc, and the opposite sides zero copper, tin and zinc. Then any point in the triangle represents one alloy, and all the possible points represent all the possible alloys of the three metals. Now divide each altitude into ten parts, and through the points of division draw lines parallel to the three sides. The crossing points of these lines represent all the alloys whose constituents are even multiples of ten per cent. We will make these alloys and determine their tensile strength, and we will cut a lot of straight wires to lengths corresponding to the strength; then we will set these wires vertically on a board which has the triangle drawn upon it, in holes drilled at the points representing the alloys. We will then fill in this forest of wires with plaster of paris, smoothing it off so as just to leave the tops of the wires visible. We will thus have a topography which shows the complete law of the relation of the tensile strength of the triple alloys to their composition." I well remember my amazement when he had completed the description, that a man with such a worn-out brain should be capable of such a brilliant piece of intellectual work and invention. The investigation of the triple alloys was carried out exactly as he had planned it, and the results were published in the Reports of the United States Test Board, and in Dr. Thurston's book on Alloys.

In 1873 Dr. Thurston discovered the phenomenon of the elevation of the elastic limit of iron and steel. It was also discovered independently in the same year by Commander L. A. Beardslee, U.S.N. In years following he tried to find other metals or alloys that exhibited the same peculiar action after being strained beyond their elastic limit, but never found one. In 1874 he investigated the burning of tan bark for fuel in specially constructed boiler furnaces, and in 1875 and 1876 he made a number of tests of steam boilers. For some years after 1876 he carried on investigations of lubricants, the results of which are in his book on Friction and Lost Work. In later years his researches were not numerous or important, for the reason that his time was fully occupied with other work. It is a matter for lasting regret that the work of the United States Iron and Steel Test Board was discontinued almost before it was fairly started, and before the Watertown testing machine, which was built for its use, was finished.

Dr. Thurston was an omniverous reader, and a tremendously active writer. Prior to 1880 most of his technical writings were contributed to the American Society of Civil Engineers and to the Journal of the Franklin Institute. After that date his engineering papers were mostly given to The American Society of Mechanical Engineers.

Dr. Thurston's literary work was not confined to engineering matters. In 1873 he contributed to the Scientific American a series of seventeen articles on his observations in Europe, which included not only what he had seen at the Vienna Exhibition, and in the several iron works that he visited, but also his reflections on social conditions in the manufacturing centers. Here is a quotation from his remarks on the inferiority of workmanship which then characterized many European productions:

> A liberalization of patent codes, and the gradual training of the workmen of Europe to a knowledge of the importance of good workmanship, and of the methods of securing it, will at a time which we hope is not far distant, do much toward the improvement of the condition of the people. We draw some of our best material from amongst them, and it seems sufficiently evident that not upon nature but upon man's own imperfect political systems lies the responsibility of the unsatisfactory condition of manufactures in Europe.

Dr. Thurston's first important book after his report of the Vienna Exhibition was his History of the Growth of the Steam Engine, published by Appleton in 1878. It is written in his best style, and to those who are at all interested in the subject it is as readable as a novel. It illustrates his painstaking care to be sure of his facts, his skill in arranging them in logical order, and his good judgment in drawing conclusions. In 1877 he brought out a little book on Steam Boiler Explosions in Theory and Practice.

In 1879 he had his second and last nervous breakdown, more serious than the first, so that he was compelled to spend more than a whole year in a sanitarium, doing no work of any kind. He returned to work in 1880, and then followed twelve years of most intense literary activity, during which he brought out an average of a book every year, some of them large octavos of 1000 pages. Here is a list of the books whose first editions appeared in the years 1882 to 1894 inclusive:

The Materials of Engineering. 3 vol. I, Non-metallic materials; II, Iron and Steel; III, Brasses, Bronzes and other Alloys.

A Text Book of Materials of Construction.

Treatise on Friction and Lost Work in Machinery and Millwork.

Manual of Steam Boilers, their Design, Construction and Operation.

Handbook of Engine and Boiler Trials, the Indicator and the Prony Brake.

Translation of Sadi Carnot's "Reflections on the Motive Power of Heat and on Machines fitted to Develop that Power."

Life of Robert Fulton.

Manual of the Steam Engine. 2 vol. I, History, Structure and Theory; II, Design, Construction and Operation.

Stationary Steam Engines, Simple and Compound.

The Animal as a Machine and Prime Motor, and the Laws of Energetics.

Many of these books ran through several editions; the second volume of Materials of Engineering, Iron and Steel, is now in its ninth and the Treatise on Friction and Lost Work, the Manual of Steam Boilers, and Stationary Steam Engines, are each in their seventh edition. The work of revising these books to keep them up to date was no small labor, and several of them have as many as four copyright dates. The Manual of the Steam Engine, and the Handbook of Engine and Boiler Trials, have been translated into French.

Most of these books are severely technical and of interest only to engineers and engineering students; but two of them , the translation of Carnot's little book and The Animal as a Machine, appeal also to those interested in advanced physics. He translated Carnot not because he thought the book would sell, but as he says in his preface, "as a matter of limited but most intense scientific interest," and he compliments the publishers for their undertaking to print the book without any prospect of financial return. Yet in seven years a second edition was printed. Dr. Thurston's appreciation of Carnot in the introduction is a good example of his literary style when writing on non-technical subjects.

Nicholas-Leonard-Sadi-Carnot was perhaps the greatest genius in the department of physical science that this century has produced. By this I mean that he possessed in the highest degree that combination of the imaginative faculty with intellectual acuteness, great logical power and capacity for learning, classifying and organizing in their proper relations all the facts, phenomena and laws of natural science, which distinguishes the real genius from other men and even from simply talented men. Only now and then in the centuries does such a man come into view. Euclid was such in mathematics, Newton was such in mechanics, Bacon and Comte were such in logic and philosophy, Lavoisier and Davy were such in chemistry, and Fourier, Thomson, Maxwell,

and Clausius were such in mathematical physics. Among engineers we have the examples of Watt as inventor and philosopher, and Rankine as his mathematical complement, developing the theory of that art of which Watt illustrated the practical side.

But Carnot exhibited that most marked characteristic of real genius, the power of applying such qualities as I have just enumerated to great purposes and with great result while still a youth. Genius is not dependent, as is talent, upon the ripening and the growth of years for its prescience; it is ready at the earliest maturity, and sometimes earlier, to exhibit its marvelous works; as for example note Hamilton, the mathematician, and Mill, the logician, the one becoming master of a dozen languages when hardly more than as many years of age, reading Newton's Principia at sixteen, and conceiving that wonderful system, quaternions, at eighteen; the other competent to begin the study of Greek at three, learning Latin at seven, and reading Plato before he was eight. Carnot had done his grandest work of the century in his province of thought and had passed into the Unseen at 36; his one little volume, which has made him immortal, was written when he was but 23 or 24.

A fine example of Dr. Thurston's grasp of a subject of scientific thought beyond the domain of engineering, and even beyond the present borderland of physics, is seen in the following brief extract from The Animal as a Machine:

The living body is a machine in which the law of Carnot, which asserts the necessity of waste in all thermodynamic processes and in every heat engine, and which shows that waste to be the greater as the range of temperature worked through by the machine is the more restricted, is evaded; it produces electricity without intermediate conversions and losses; it obtains heat without high temperature combustion; and in some cases light without any sensible heat. In other words, in the vital system of man and of the lower animals, nature shows us the practicability of converting any one form of energy into any other, without those losses and unavoidable wastes characteristic of the methods, the invention of which has been the pride and the boast of man. Every living creature, man and worm alike, shows him that his task is but half accomplished; that his grandest inventions are but crude and remote imitations; that his best work is wasteful and awkward. Every animate creature is a machine of enormously higher efficiency as a dynamic engine than his most elaborate constructions. Every gymnotus living in the mud in tropical stream puts to shame man's best effort in the production of electricity; and the minute insect that flashes across his lawn on a summer evening, or the worm that lights his path in the garden, exhibits a system of illumination incomparably superior to his most perfect electric lights . . . Here is Nature's challenge to man. Man wastes one-fourth of all his fuel as utilized in his steam boiler, and often 90 per cent as used in his fireplace; nature in the animal system utilizes substantially all.

Dr. Thurston was an occasional contributor to such journals as the Popular Science Monthly, the Forum, Science, and the North American Review. Sometimes he went outside of the field of the

physical sciences and wrote on sociological and economic subjects, such as the tariff. Once he wrote for the North American Review a statement of his religious convictions.

He frequently wrote papers and delivered addresses on educational subjects, and in these he was naturally at his best. For nearly forty years he was an educator and an educational leader. He was versed in the theories of Froebel, Milton and Comenius; and of Spencer, John Scott Russell and other modern writers. His paper on Technical Education in the United States; its Social, Industrial and Economic Relations to our Progress, read at the International Engineering Congress at Chicago in 1893, is one of his masterpieces. It is a calm, scholarlike review of the conditions of the past, and a hopeful view of the future. He propounded no new theories, he originated no new fads, but he was in line with the best thinkers of his time, and thoroughly in sympathy with the modern trend toward industrial or trade education for the great mass of the people.

DR. THURSTON AT SIBLEY COLLEGE, CORNELL UNIVERSITY

By Walter C. Kerr, Mem, Am.Soc.M.E., Trustee Cornell University

What a man is, makes less difference to the world than what his life teaches; the man departs, but his teaching remains. It would be impossible for me to relate here the full importance of the eighteen years that Dr. Thurston devoted to Cornell University. Prior to 1885 Cornell was developing a department of mechanic arts in a small way, recognizing the necessity and opportunities of mechanical engineering. With a profitable sale of the university's timber lands the trustees felt warranted in taking forward steps, chief among which was the founding of a new department, and to this department of mechanical engineering Dr. Thurston was called as the first director. No choice was ever more fortunate. I will not undertake to recount all that followed in physical development from his administration, except to say that the number of students increased from one hundred to eleven hundred, buildings grew, facilities grew, everything that his hand touched grew, and all the growth was healthy. Professor Thurston was especially an organizer, and of the very best kind. This was because he knew what to organize. His methods were direct and practical, he knew men, he understood human nature; and all resistance was to him merely a retardation, not a stopping, and consequently he gained whatever he set out to do.

By temperament, education and experience he was peculiarly fitted to direct socially and intellectually an important department in a complex institution; by his touch with all the forces of life he was an important factor in any community in which he lived, and this gave him a profound and wide influence for good through a much larger circle than that of engineering. He convinced men, by persuading them to want what he wanted, and the result was that he usually gained his end with the minimum of argument. His ever-present cheerfulness was an inspiration, and his patience was an example. There is no subtle mystery about why he was so loved and respected at Cornell, nor why he accomplished so much. His ways were ways of peace, and his achievements were a series of creative victories. He was a strong man, so strong that we honor his memory tonight. He has gone, but the influence of his life lives.

TEST OF A 15,000-KW. STEAM-ENGINE-TURBINE UNIT

H. G. Stott, New York
Member of the Society

R. J. S. Pigott,[1] New York
Non-Member

During the year 1908 it became apparent that owing to the cost increasing traffic in the New York subway, it would be necessary to have additional power available for the winter of 1909–1910.

2 The power plant of the Interborough Rapid Transit Company, which supplies the subway, is located on the block bounded by 58th and 59th Streets, and by 11th and 12th Avenues, adjacent to the North River; it contains nine 7500-kw. (maximum rating) engine units, besides three 1250-kw. 60-cycle turbine units which are used exclusively for lighting and signal purposes.

3 The 7500-kw. units consist of Manhattan-type compound Corliss engines, having two 42-in. horizontal high-pressure cylinders and two 86-in. vertical low-pressure cylinders. Each horizontal high-pressure cylinder and vertical low-pressure cylinder has its connecting rod attached to the same crank, so that the unit becomes a four-cylinder 60-in. stroke compound engine with an overhanging crank on each side of a 7500-kw. maximum rating 11,000-volt, three-phase, 25-cycle generator. The generator revolving field is built up of riveted steel plates of sufficient weight to act as a flywheel for the two engines connected to it. This arrangement gives a very compact two-bearing unit. The valve gear on the high-pressure cylinders is of the poppet type, and on the low-pressure of the Corliss double-ported type.

4 The condensing apparatus consists of barometric condensers,

[1]Interborough Rapid Transit Company.

The American Society of Mechanical Engineers, 29 W. 39th St., N. Y.
All papers are subject to revision.

arranged so as to be directly attached to the low-pressure exhaust nozzles, with the usual compound displacement circulating pump and simple dry-vacuum pump.

5 These engine and generator units are in general probably the most satisfactory large units ever built, as five years' experience with them has proved; their normal economic rating is 5000 kw., but they operate equally well (water rate excepted) on 8000 kw. continuously.

6 In considering the problem of how to get an additional supply of power, every available source was considered, but by a process of elimination only two distinct plans were left in the field.

7 The electric transmission of power from a hydraulic plant was first considered, but owing to the high cost of a double transmission line from the nearest available water power, and the impossibility of getting reliable service (that is, service having a maximum total interruption of not more than ten minutes per annum) from such a line, further consideration of this plan was abandoned.

8 The gas engine, while offering the highest thermo-dynamic efficiency at the same time required an investment of at least 35 per cent more than an ordinary steam-turbine plant with a probable maintenance and operation account of from four to ten times that of the steam turbine.

9 The reciprocating-engine unit of the same type as those already installed, was rejected in spite of its most satisfactory performance, on account of the high first cost and small range of economical operation. Reference to Fig. 1, Series A will show that the economic limits of operation are between 3300 kw. and 6300 kw.; beyond these limits the water rate rises so rapidly as to make operation undesirable under this condition, except for a short period during peak loads.

10 The choice was thus narrowed down to either the high-pressure steam turbine or the low-pressure steam turbine. There was sufficient space in the present building to accommodate three 7500-kw. units of the high-pressure type, or a low-pressure unit of the same size on each of the nine engines, so that the questions of real estate and building were eliminated from the problem.

11 The first cost of a low-pressure turbine unit is slightly lower than that of a high-pressure unit, due to the omission of the high pressure stages and the hydraulic governing apparatus, but the cost of the condensing apparatus would be the same in both cases. The foundations and the steam piping in both cases would not differ greatly. The economic results, so far as the first cost is concerned, would then be approximately the same, if we consider the general

case only; but in this particular instance the installation of high-pressure turbines would have meant a much greater investment for foundations, flooring, switchboard apparatus, steam piping and water tunnels, amounting to an addition of not less than twenty-five per cent to the first cost.

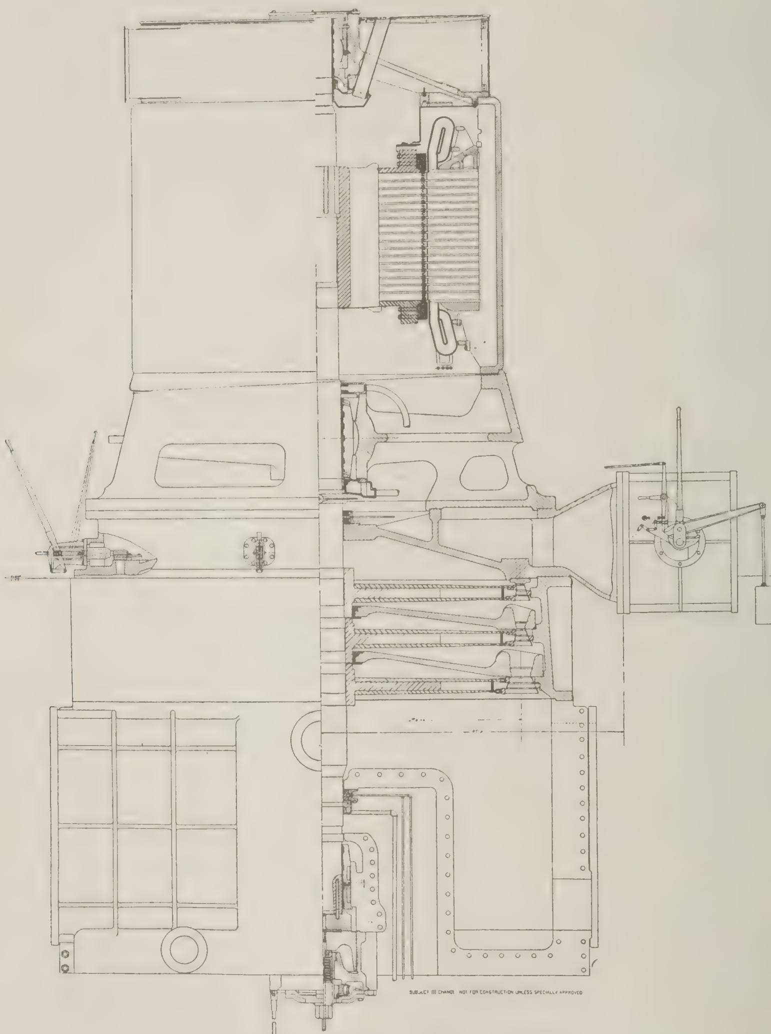
12 The general case of displacing reciprocating engines and installing steam-turbine units in their place was also considered. The best type of high-pressure turbine plant has a thermal efficiency approximately 10 per cent better than the best reciprocating-engine plant, but the items of labor for operation and for maintenance, together with the saving of about 85 per cent of the water for boiler-feed purposes and the 10 per cent of coal, reduce the relative operating and maintenance charges for the steam-turbine plant to 80 per cent, as compared to 100 per cent for the reciprocating-engine plant.

13 Assuming that the reciprocating engine plant is a first-class one and has been well maintained, about 20 per cent of its original cost (for engines, generators and condensers) may be realized on the old plant and so credited to the cost of the high-pressure turbine plant. But on the other hand, if the high-pressure turbine installation is to receive credit for the second-hand value of the engines, it must also have a debit charge for 100 per cent of the original reciprocating-engine plant which it displaced. The relative investments, therefore, upon this basis would be approximately equal for the high-pressure or the low-pressure turbine; but 80 per cent of the cost of the original engine plant would have to be charged against the high-pressure turbine plant, as against an actual increase in value (to the owner) of the engine by reason of its improved thermal efficiency, due to the addition of the low-pressure turbine.

14 The preliminary calculations, based upon the manufacturers' guarantees for the low-pressure and high-pressure turbines, showed that the combined engine-turbine unit would give at least 8 per cent better efficiency than the high-pressure turbine unit, so that it was finally decided to place an order for one 7500-kw. (maximum rating) unit, as by this means we would not only get an increase of 100 per cent in capacity, but at the same time give the engines a new lease of life by bringing them up to a thermal efficiency higher than that attained by any other type of steam plant.

15 The turbine installed is of the vertical three-stage impulse type having six fixed nozzles and six which can be operated by hand, so as to control the back pressure on the engine, or the division of load between engine and turbine. An emergency overspeed governor,

which trips a 40-in. butterfly valve on the steam pipe connecting the separator and the turbine and at the same time the 8-in. vacuum



Elevation and Part Section of Low-Pressure Turbine Unit

breaker on the condenser, is the only form of governor used. The footstep bearing, carrying the weight of the turbine and generator rotors, is of the usual design supplied with oil under a pressure of

600 lb. per sq. in. with the usual double system of supply and accumulator to regulate the pressure and speed of the oil pumps.

16 The condenser contains approximately 25,000 sq. ft. of cooling surface arranged in the double two-pass system of water circulation with a 30-in. centrifugal circulating pump having a maximum capacity of 30,000 gal. per hr. The dry vacuum pump is of the single-stage type, 12-in. and 29-in. by 24-in., fitted with Corliss valves on the air cylinder. The whole condensing plant is capable of maintaining a vacuum within 1.1 in of the barometer when condensing 150,000 lb. of steam per hr. when supplied with circulating water at 70 deg. fahr.

17 The electric generator is of the three-phase induction type, star-wound for 11,000 volts, 25 cycles and a speed of 750 r.p.m. The rotor is of the squirrel-cage type with bar winding connecting into common bus-bar straps at each end. This type of generator was chosen as being specially suited to the conditions obtaining in the plant.

18 With nine units operating in multiple, each one capable of giving out 15,000 kw. for a short tine, operating in multiple with another plant of the same size, it is evident that it is quite possible to concentrate 270,000 kw. on a short circuit. If we proceed to add to this, synchronous turbine units of 7500-kw. capacity, which, owing to their inherently better regulation and enormous stored energy, are capable of giving out at least six times their maximum rated capacity, the situation might soon become dangerous to operate, as it would be impossible to design switching apparatus which could successfully handle this amount of energy. The induction generator, on the other hand, is entirely dependent upon the synchronous apparatus for its excitation, and in case of a short circuit on the bus-bars would automatically lose its excitation by the fall in potential on the synchronous apparatus.

19 The absence of fields leads to the simplest possible switching apparatus, as the induction generator leads are tied in solidly through knife switches, which are never opened, to the main generator leads. The switchboard operator has no control whatever over the induction generator, and only knows it is present by the increased output on the engine generator instruments.

20 The method of starting is simplicity itself—the exciting current is put on the engine generator *before* starting the engine, and then the engine is started, brought up to speed and synchronized in exactly the same way as before. While starting in this way, the induction

generator acts as a motor until sufficient steam passes through the engine to carry the turbine above synchronism, when it immediately becomes a generator and picks up the load. Three of these 7500-kw. low-pressure turbine units have been installed and tests run on Nos. 1 and 2. No. 3, having been just started, has not yet been tested.

21 Instead of inserting in this paper the enormous accumulation of data incident to these tests, we have divided the paper into two parts in the hope that it would thus be more accessible for reference, the first part giving the reasons for adopting this particular type of apparatus, with a brief description of the plant and a summary of the results obtained, and the second part containing all the principal data acquired during the tests, with sufficient explanation to make their meaning clear without reference to the text.

22 The tables and curve sheets are as follows:

Series A: Engine tests made in connection with acceptance tests, and also later to determine best conditions for operation.

Series B: Calculations and data furnished by turbine manufacturer to determine probable results when combined with engine data obtained in Series A.

Series C: Tests on No. 1 combined unit. This unit was hurriedly put into commission in order to obtain results to determine future developments. To get the piping done, old riveted steel pipe was used which was very leaky under vacuum. Results are valuable however as showing the effect of vacuum on performance as compared to Series E and F. Quality of steam entering turbine also poor.

Series D: Tests of No. 2 unit, with poor vacuum and poor quality of steam entering turbine.

Series E and F: Tests on No. 2 combined unit; conditions of vacuum and quality of steam entering turbine nearly standard, so that corrections are small.

23 In all results, except where specially noted, moisture corrections are simple corrections, i. e., for each per cent of moisture only one per cent correction has been made. Vacuum corrections for the combined unit are 1 lb. for each inch variation from 28.5 in. when referred to 29.92 in. barometer.

24 The net results obtained by the installation of low-pressure turbine units may be summarized as follows:

a An increase of 100 per cent in maximum capacity of plant.
b An increase of 146 per cent in economic capacity of plant.
c A saving of approximately 85 per cent of the condensed steam for return to the boilers.
d An average improvement in economy of 13 per cent over the best high-pressure turbine results.
e An average improvement in economy of 25 per cent (between the limits of 7000 kw. and 15,000 kw.) over the results obtained by the engine units alone.
f An average unit thermal efficiency between the limits of 6500 kw. and 15,500 kw. of 20.6 per cent.

NUMBER OF TEST	ENG. LOAD K.W.	STEAM PR. Gauge Lbs.G.	ST'M TEMP. °F	ST'M SUPER-HEAT °F	RECEIVER PR Gauge (from Temp) Lbs.G.	VACUUM Std. 29.92" Bar. "Hg.	QUALITY STM %	WATER PER HR Lbs.	DRY STM. P. HR. Lbs.	REC. DRAIN P HR Lbs.	STM. TO AUX'L Lbs.	INJECT WATER TEMP. °F.	DISCH. WATER TEMP. °F.	I.H.P. H.P. H.P.	I.H.P. L.P. H.P.	I.H.P. TOTAL H.P.	DRY STM. p.K.W.H. Lbs.	DRY STM. per I.H.P.H. Lbs.
25	3100	180.1	388.3	9.7	9.13	28.81	100.55	56040	56349	2989	1517	36.8	55.7	2173	2306	4479	18.18	12.58
22	4008	176.7	383.9	5.7	16.87	27.99	100.32	68190	68407	4882	1866	38.4	58.3	2699	2815	5514	17.07	12.42
24	4977	174.4	387.8	10.6	21.7	28.00	100.58	85369	85865	5273	2949	36.8	65.3	3264	4076	7341	17.25	11.70
21	5984	173.9	387.5	10.5	25.9	28.00	100.60	103896	104519	6031	1699	37.73	69.5	3717	4714	8431	17.47	12.40
23	6772	173.3	385.5	8.7	30.0	27.71	100.50	124702	125326	6060	2091	37.7	70.4	4346	5732	10078	18.51	12.37
27	4992	173.9	387.3	10.5	10.44	28.11	100.60	89525	90062	5367	1826	37.76	72.66	3770	5184	6954	18.04	12.95
26	4970	173.2	386.1	9.4	15.21	28.00	100.53	86267	86724	5518	1728	35.9	61.1	3443	3452	6895	17.45	12.58
29	4976	174.9	386.9	9.4	20.41	28.00	100.53	85557	86010	5294	3034	38.1	75.0	3124	3722	6846	17.29	12.59
28	4970	174.3	388.8	11.7	25.35	28.02	100.66	85933	86501	4890	663	36.7	72.6	2982	4127	7109	17.42	12.17
31	3988	177.7	387.5	8.9	32.62	—	100.51	109317	109874	5948				2625	2372	4997	27.55	21.99
32	4980	176.2	386.7	8.7	36.93	—	100.50	128056	128695	6352				2835	3333	6168	25.84	20.88
30	4961	148.2	372.0	7.0	21.06	28.04	100.40	88041	88394	5082	1284	37.46	72.4	3068	4019	7087	17.82	12.48

TABLE 1 SERIES A, ENGINE TESTS

NUMBER OF TEST	ENG. LOAD K.W.	B.T.U. ADDED PER POUND WATER	B.T.U. REJ. PER POUND WATER	EFF. RANKINE %	EFF. THERMAL %	$\frac{EFF_T}{EFF_R}$ %	B.T.U. DRAINS %	B.T.U. CONS'R & RAD'N LOSS %	B.T.U. MECH. ELEC. LOSS %	REMARKS
25	3100	1205	840	30.3	15.7	51.7	0.9	71.4	12.0	
22	4008	1202	865	28.0	16.7	59.6	1.3	70.0	Do.	
24	4977	1204	866	28.1	16.5	58.1	1.2	70.3	Do.	
21	5984	1204	866	27.1	16.3	58.2	1.1	70.6	Do.	
23	6772	1203	875	28.3	15.4	56.4	1.0	71.6	Do.	
27	4992	1205	865	28.2	15.8	56.0	1.0	71.2	Do.	
26	4970	1204	866	28.1	16.3	58.1	1.2	70.5	Do.	
29	4976	1205	866	28.1	16.4	58.6	1.2	70.4	Do.	
28	4970	1206	866	28.2	16.4	58.1	1.1	70.5	Do.	
31	3988	1205	1017	15.6	10.3	66.2	1.1	76.6	Do.	Non-condensing
32	4980	1204	1017	15.5	11.0	71.1	1.0	76.0	Do.	Non-condensing
30	4961	1200	875	27.1	16.0	59.7	1.1	70.9	Do.	

TABLE 2 SERIES A

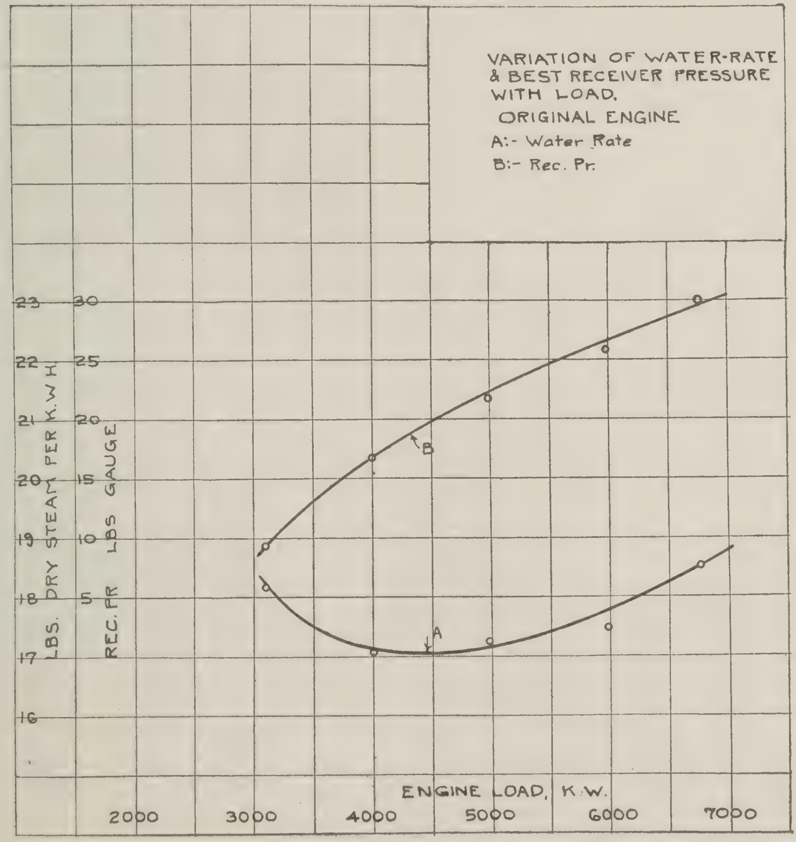


FIG. 1 SERIES A

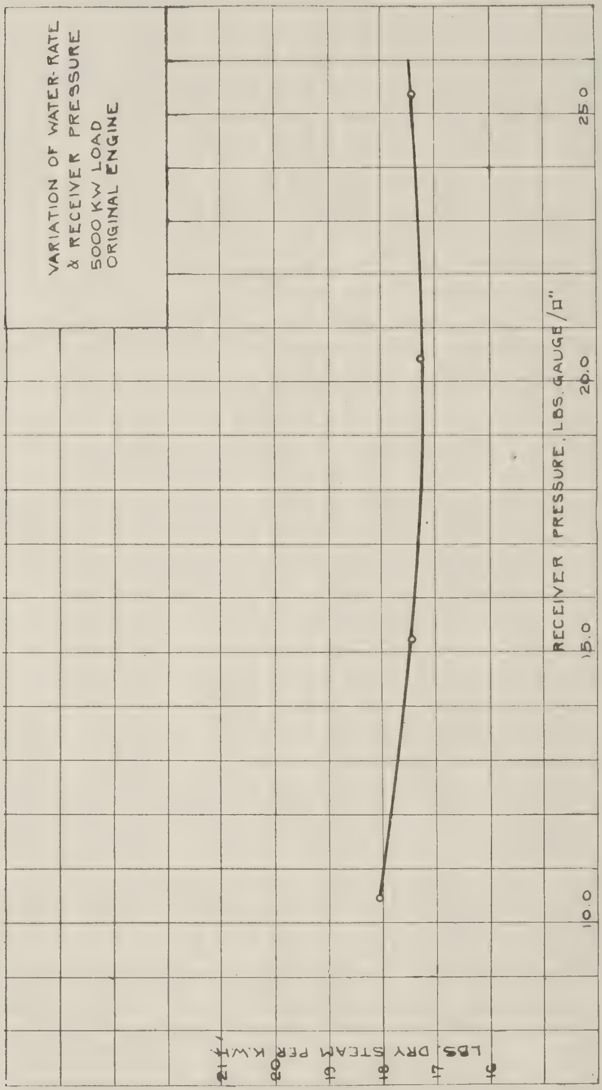


FIG. 2 SERIES A

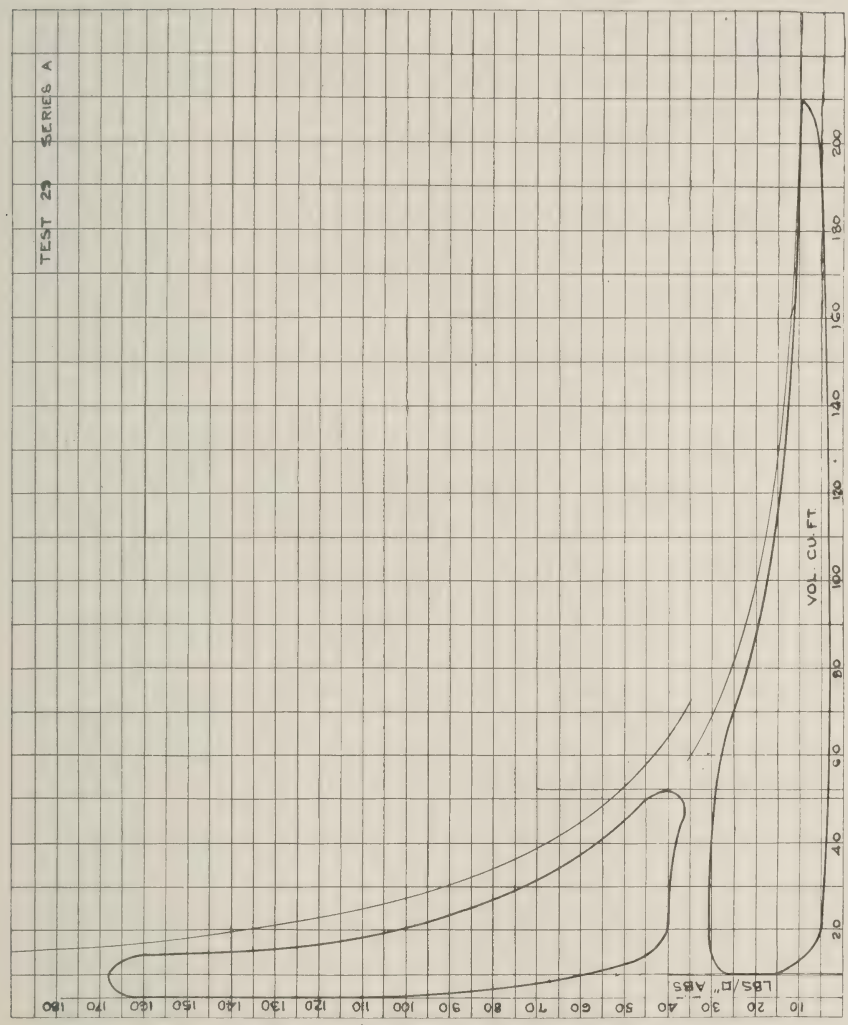


Fig. 3 Series A, Test 29

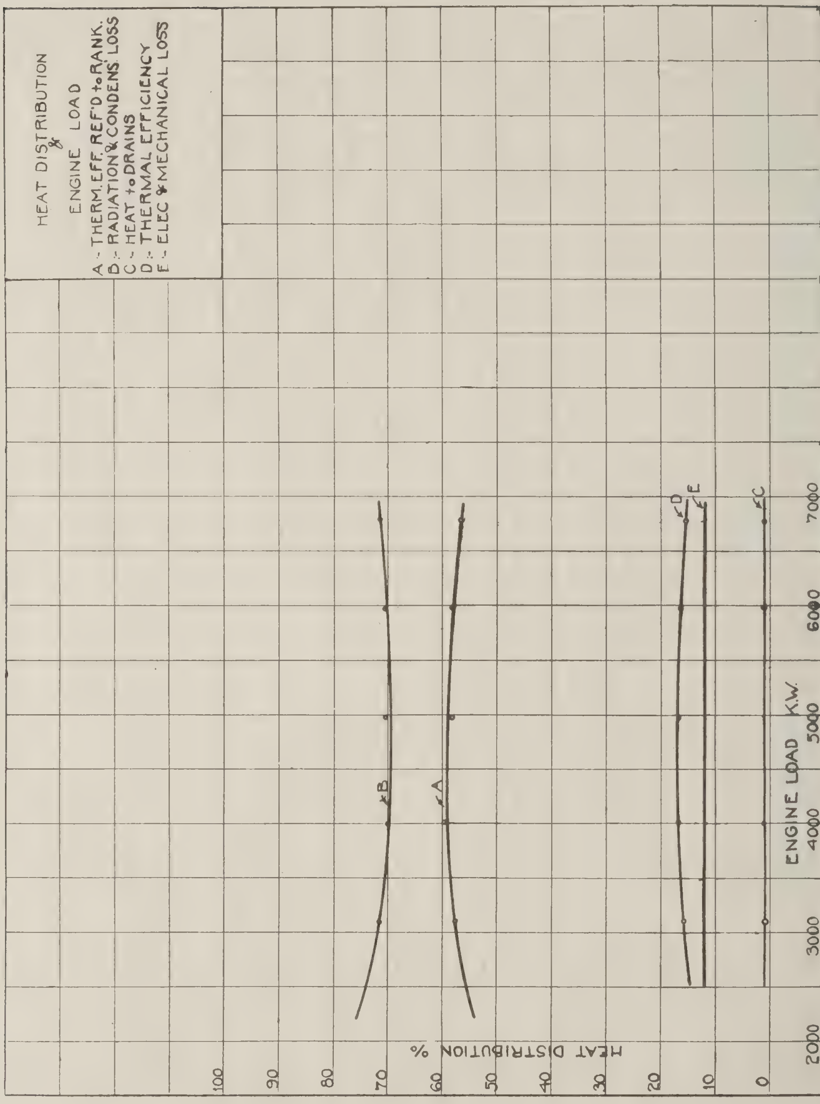


FIG. 3a SERIES A

NO TEST	LOAD	I.H.P	RATIO HP/LP I.H.P	p_a	p_c	V_a	V_c	w_a	w_c	STM. PER STROKE H.P. CARD	r	y	INDIC. WATER RATE per I.H.P.	ACTUAL WATER RATE per I.H.P.	ACTUAL WATER RATE per K.W.H.	RECEIVER PRESS.	EXHAUST PRESS.	STM. PRESS.	TOTAL WATER
.	KW	IHP		Lbs/□" Abs.	Lbs/□" Abs.	Cu. Ft.	Cu. Ft.	Lbs/Cu Ft	Lbs/Cu Ft	Lbs.			Lbs/Hr.	Lbs/Hr.	Lbs/Hr	Lbs/□" Abs.	Lbs/□" Abs.	Lbs/□" Abs.	Lbs/Hr
25	3100	4479	.943	137.32	66.12	10.78	7.56	.3058	.1544	2.131	4.80	.469	8.57	12.59	18.18	19.8	.9	191.5	56349
22	4008	5515	.959	143.0	65.3	13.77	9.31	.3176	.1526	2.951	3.41	.298	9.64	12.41	17.07	31.6	1.0	192.8	68407
29	4976	6846	.840	142.4	73.8	16.64	9.30	.3163	.1709	3.670	3.11	.302	9.65	12.56	17.29	36.4	1.0	190.9	86010
24	4977	7341	.801	141.8	75.7	17.13	9.30	.3151	.1751	3.769	3.32	.265	9.24	11.69	17.25	35.1	.9	191.0	85865
21	5984	8431	.789	141.8	74.1	21.48	9.29	.3142	.1716	5.154	2.42	.128	11.00	12.40	17.47	40.6	.9	191.0	104519
23	6772	10078	.759	140.6	89.0	25.40	9.29	.3126	.2037	6.080	2.04	.154	10.86	12.44	18.51	44.7	1.0	190.9	125326
31	3988	5900	1.403	143.7	72.2	19.83	7.56	.3191	.1675	5.058	3.11	.206	15.43	18.62	27.55	47.3	14.7	192.2	109874
32	4981	7004	.807	143.7	75.8	23.44	7.57	.3191	.1753	6.154	2.21	.155	15.81	18.28	25.84	51.6	14.7	189.7	128695
ASSUMED CARDS, VARIABLE NOZZLE PRESSURE																			
A	4050	5936	1.04	187.8	82.5	10.93	4.70	.4107	.1897	3.597	4.73	.474	12.00	17.69	25.92	39.7	7.0	218	105000
B	5590	8192	1.05	182.2	96.0	16.45	"	.3991	.2186	5.537	3.14	.268	12.18	15.45	22.63	46.0	10.0	"	126600
C	6710	9836	1.07	176.0	119.5	22.80	"	.3862	.2685	7.55	2.27	.156	13.62	15.98	23.40	52.2	13.5	"	157300
D	7370	10792	1.10	168.6	122.8	29.83	"	.3708	.2753	9.71	1.73	.087	16.30	17.72	25.95	58.5	18.0	"	191100
E	7740	11336	1.05	160.9	136.2	37.60	"	.3549	.3034	11.92	1.31	.033	18.91	19.54	28.61	64.7	23.0	"	221400
ASSUMED CARDS, CONSTANT NOZZLE PRESSURE																			
F	3875	5676	1.12	182.2	136.2	16.45	"	.3991	.3034	5.140	3.14	.268	16.31	20.69	30.30	64.7	16.0	"	117400
G	5795	8484	.988	176.0	"	22.80	"	.3862	"	7.374	2.27	.156	15.57	18.00	26.38	"	"	"	152700
H	7190	10540	.974	168.6	"	29.83	"	.3708	"	9.633	1.73	.087	16.45	17.88	26.20	"	"	"	188500
I	8200	12008	.998	160.9	"	37.60	"	.3549	"	11.92	1.31	.033	17.86	18.45	27.04	"	"	"	221700

TABLE 3 SERIES B

ASSUMED CARDS L.P EXHAUST QUALITY DATA.

NO	WATER p.Hr. Lbs.	H.P. STM. TO L.P. CYL %	MOISTURE at L.P. Adm %	ADM PR L P Abs Lbs/□"	REL PR. L. P Abs Lbs/□"	EXH PR L P Abs Lbs/□"	r	QUAL OF L.P. EXHAUST %	COMB. QUAL %	DRY STM. TURB. Lbs/Hr.	
A	105000	93.2	2.5	37	9.5		2.76	90.6	84.4	88600	VNP
B	126600	94.3	3.0	43	14.		2.47	90.9	85.7	108500	"
C	157300	95.2	3.5	49	19		2.26	91.4	86.9	136700	"
D	191100	95.9	4.0	55	24		2.10	91.6	87.8	167800	"
E	221400	96.2	4.0	60	28		1.98	91.9	88.4	195600	"
F	117400	96.8	3.0	60	20	17.5	3.30	90.8	84.5	99200	CNP
G	152700	95.7	3.5	"	20	"	2.94	90.5	85.5	130600	"
H	188500	94.4	4.0	"	23	"	2.46	90.8	86.9	163800	"
I	221700	93.1	4.0	"	27	"	1.98	91.6	88.7	196600	"

REMARKS & FORMULAE

TESTS 21–29 INCLUSIVE, 8 HRS.

TESTS 31–32, 8 HRS. ATMOSPHERIC EXHAUST NON-CONDENSING

$$\frac{IHP}{KW} = 1.465$$

$r = \frac{51.7}{V_a}$ for HP Card = Ratio of Expansion

$y = 0.129\,(r - 1.06)$ = Missing Water

ω = Sp. density @ p

$$W_a \times V_a = \overline{W_1} \qquad W_c \times V_c = \overline{W_2} \qquad \overline{W_1} - \overline{W_2} = \overline{W_3}$$

$$\frac{\overline{W_3} \times 60 \times 4 \times 75}{IHP} = IWR \text{ @ HP Cut-off}$$

$$IWR \times (1 + Y) = AWR/IHP/HR.$$

TABLE 4 SERIES B

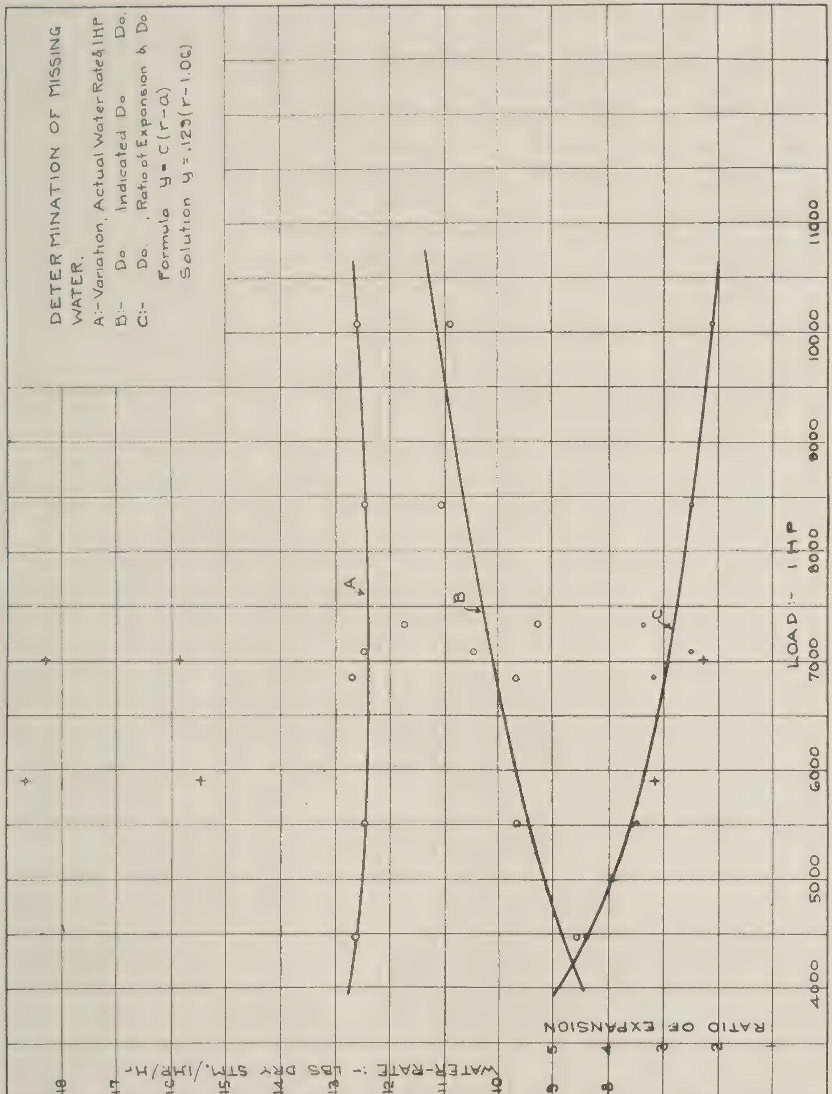


Fig. 4 Series B

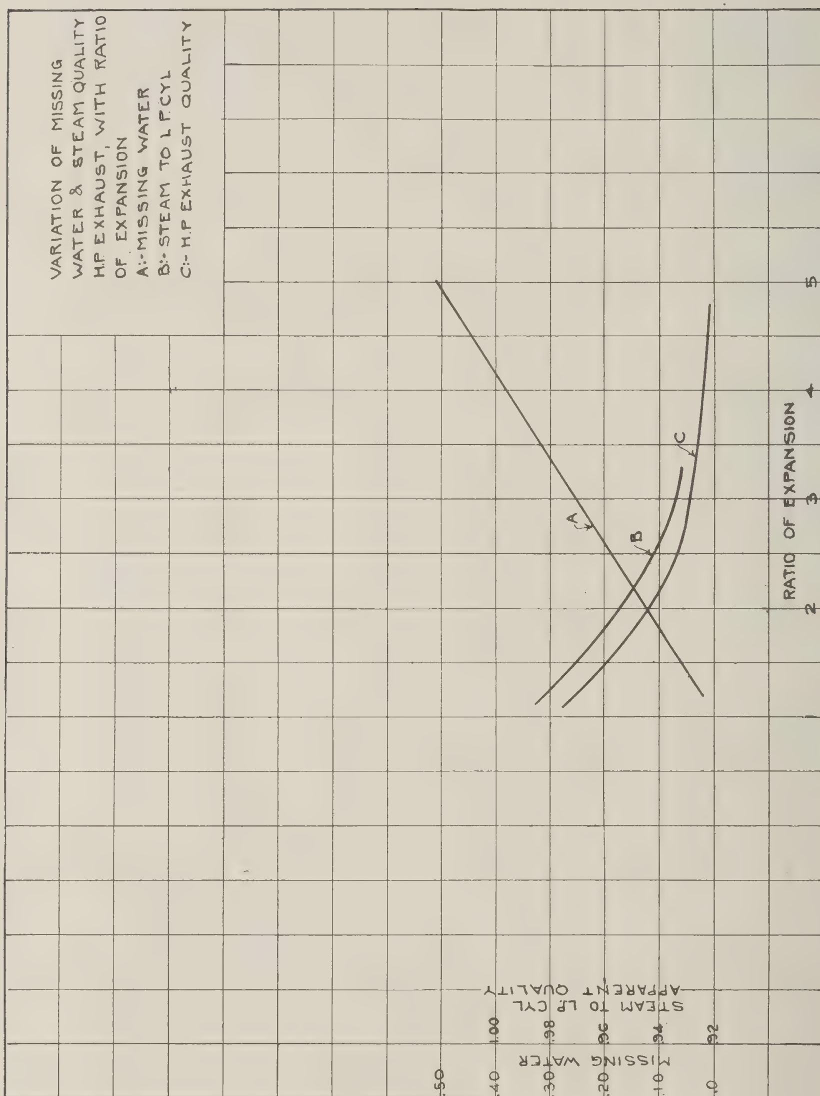


FIG. 5 SERIES B

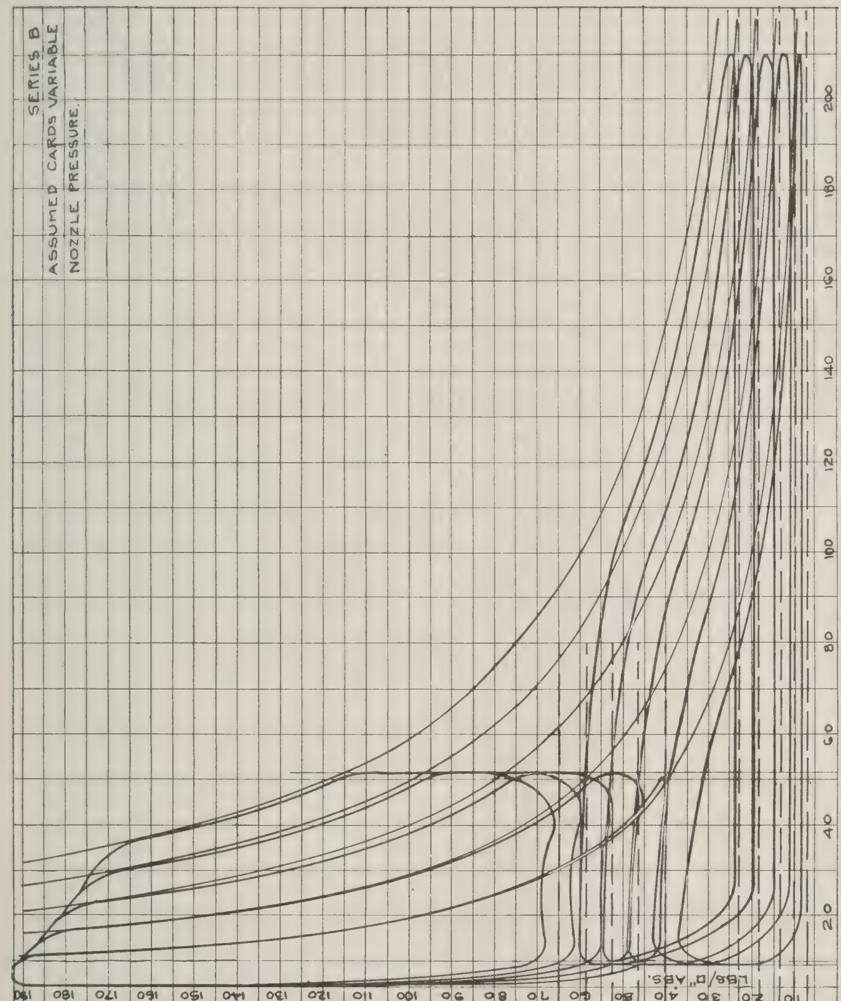


FIG. 6 SERIES B

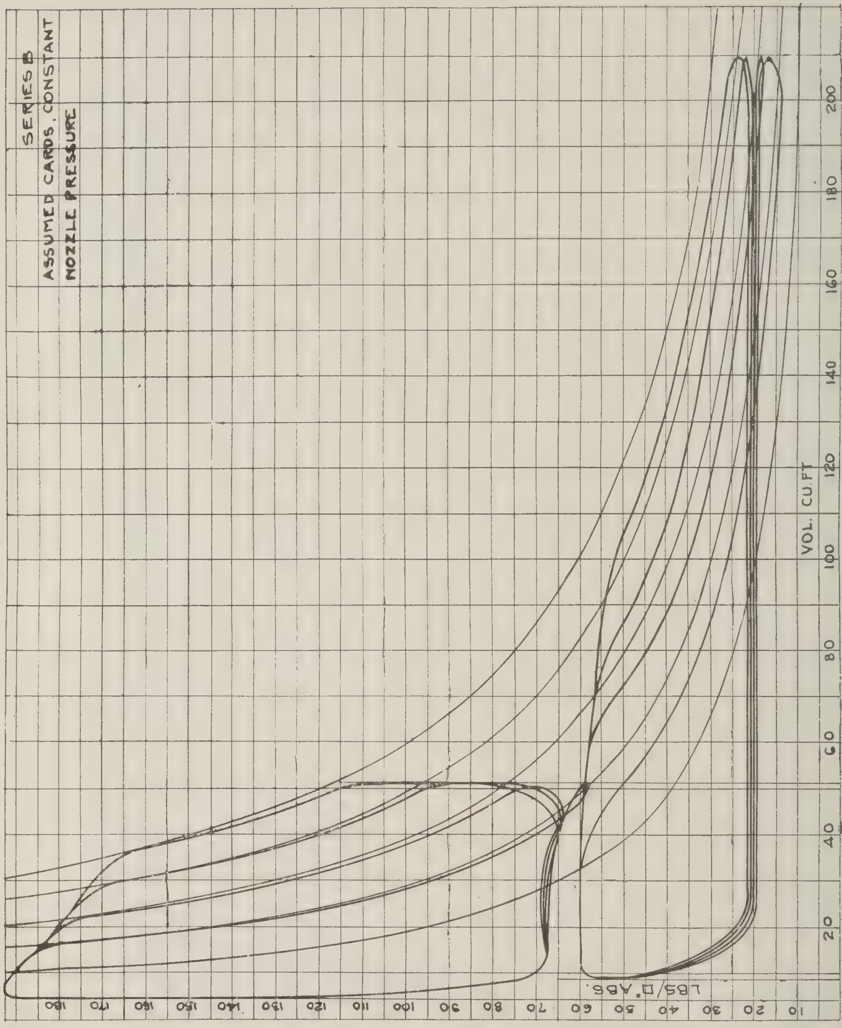


FIG. 7 SERIES B

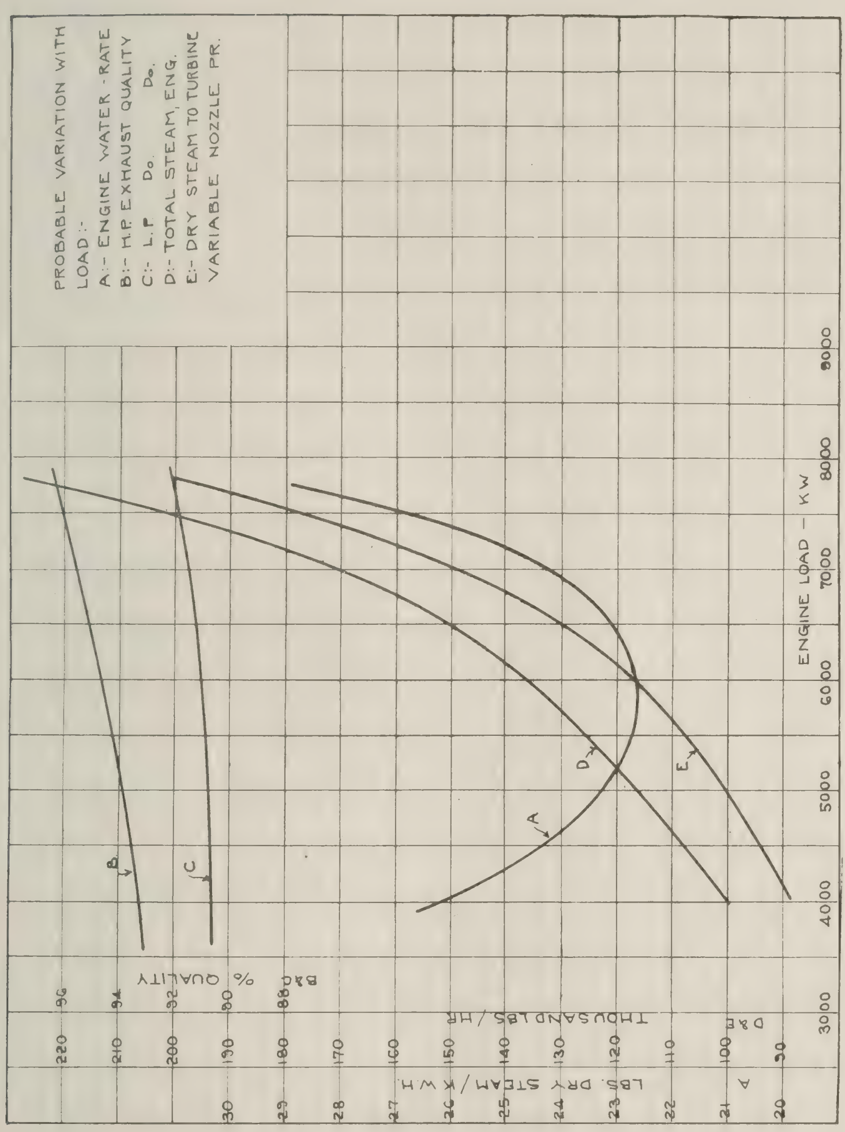


FIG. 8 SERIES B

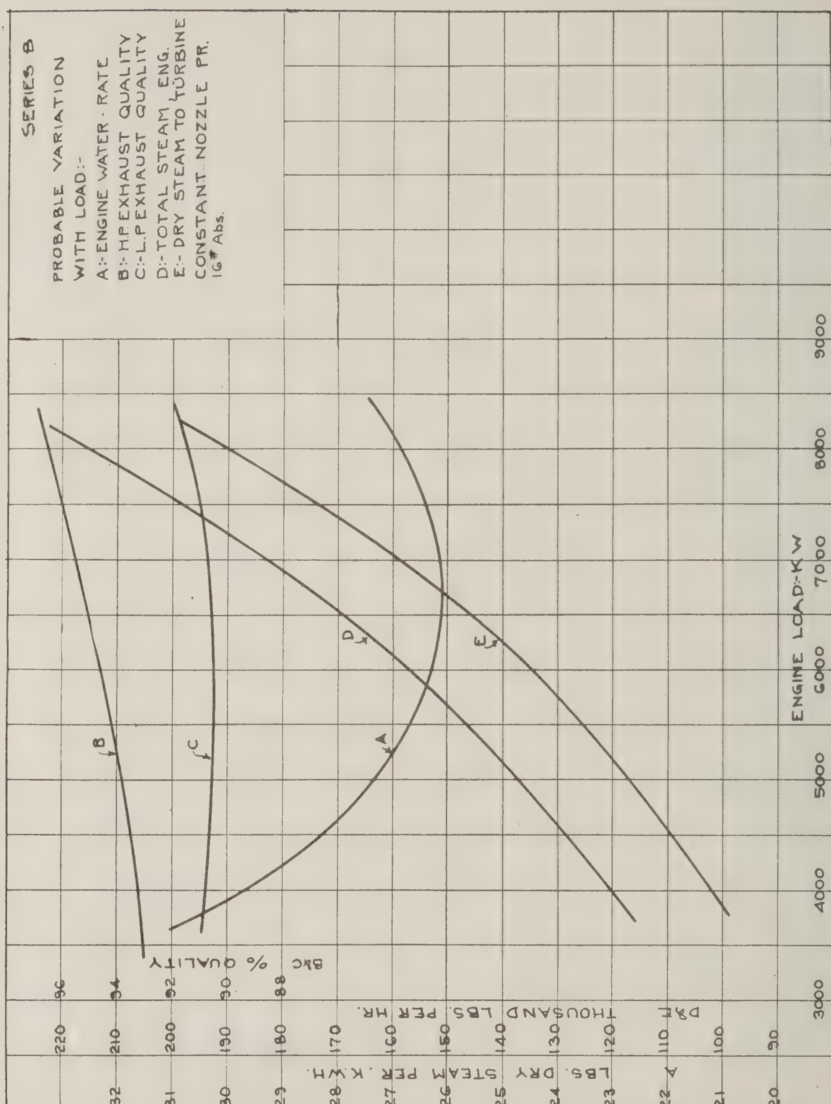


FIG. 9 SERIES B

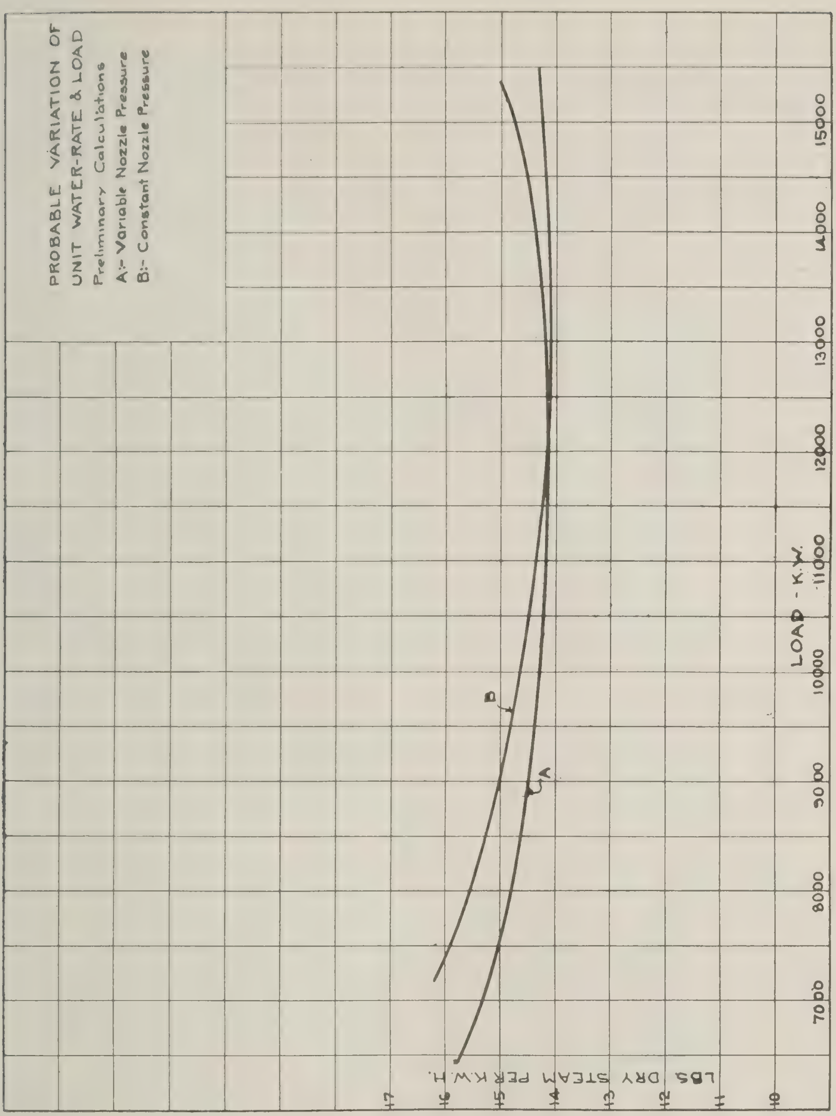


FIG. 10 SERIES B

RESUMÉ OF OBSERVATIONS										CORRECTED TO 29.92" BAR. & 14.7# ATMOS.									
TEST NUMBER	DURATION	LOADS			PRESSURES					QUALITIES		WATER			DRY STEAM		TEMPERATURES		
		UNIT	ENGINE	TURBINE	H.P. STEAM	RECEIVERS	L.P. STEAM	L.P. STEAM	VACUUM	H.P. STEAM	L.P. TURB	SEPARATOR L.P.	TURBINE DISCH.	TOTAL UNIT	TURBINE	ENG. & UNIT	HOT WELL COND	INJECTION	DISCH CIRC WATER
	Hours.	K.W.	K.W.	K.W.	Lbs. Gauge p.□"	Lbs. Gauge p.□"	Lbs Abs. p.□"	Lbs. Gauge p.□"	Ins. Hg.	%	%	Lbs. per Hour	Lbs. per Hour	Lbs. per Hour	Lbs. per Hour	Lbs. per Hour	°F	°F	°F
5	4	10220	6627	3690	174.4	36.2	12.72	—	28.03	97.08	90.89	3369	150170	153540	136500	149000	88.1	72.1	84.0
6	5	11320	6520	4870	173.4	35.7	17.22	1.52	28.34	96.96	94.09	2759	167190	169950	158200	164850	87.6	74.7	87.8
7	3	11150	6187	4960	173.4	33.9	17.36	2.66	28.55	97.06	90.17	1157	164945	166100	148350	161200	85.5	72.5	86.0
8	3	10970	6178	4840	170.2	49.6	17.36	2.66	28.44	97.15	88.62	654	161612	162270	143200	157700	86.9	74.4	86.6
9	2	11250	6560	4685	170.2	50.1	16.61	1.91	28.18	97.08	92.08	5830	164458	170290	151900	165300	87.8	72.8	86.3
10	1	12440	7060	5400	172.0	49.3	17.60	2.90	27.91	97.54	86.96	3133	185100	188230	161400	183600	89.5	72.0	83.3
11	2	8990	5195	3795	172.0	50.0	16.61	1.91	27.92	97.54	91.32	469	136850	137320	124890	133930	87.3	72.4	83.5
12	1	13240	7580	5610	172.6	48.8	17.06	2.36	27.99	98.32	85.06	6018	192290	198310	163500	194900	90.7	75.4	89.2
13	2	10240	6070	4220	172.0	47.1	16.80	2.10	27.98	98.50	91.05	825	156470	157290	142470	154900	88.2	72.8	83.8
14	2	11480	6140	5340	174.8	51.4	18.84	4.14	28.29	98.41	90.24	337	176210	176550	159010	173740	86.4	73.0	86.6
15	5	11504	6022	5468	170.8	48.1	19.45	4.75	27.51	97.83	95.15	—	167103	—	159000	—	91.8	78.2	93.5
16	7	11526	6314	5203	173.9	48.4	19.45	4.75	27.42	98.02	93.57	9633	167085	176718	157300	173200	92.9	78.6	91.5
17	5	11528	6084	5488	175.7	49.5	19.90	5.20	27.62	97.70	94.68	5111	179170	184301	169650	180100	86.4	73.6	86.7
18	6	10740	5916	4831	174.1	49.5	18.56	3.86	27.65	98.19	94.83	12554	149148	161702	141450	158750	89.7	72.8	83.6
19	6	14540	7860	6692	174.0	51.7	18.42	3.72	27.78	97.66	95.07	16404	201351	217755	191450	212700	90.9	72.2	88.2
20	4	14365	7373	6962	171.6	52.2	20.94	6.24	27.75	97.88	96.47	15423	196680	212103	189750	207750	90.3	72.6	93.7
21	6	10320	5632	4617	175.9	49.8	17.77	3.07	28.03	97.54	89.52	6486	144399	150885	129150	147100	87.8	72.5	84.4
23	6	13410	7460	5862	173.9	50.2	18.01	3.31	27.72	97.80	96.09	15760	195514	211274	188000	206800	91.1	73.0	87.0
24	6	12927	7033	5820	169.8	50.2	18.59	3.89	27.79	98.03	95.47	14439	173150	187589	165300	183950	89.8	73.3	86.8
25	6	9730	5142	4508	175.3	49.7	18.99	4.29	27.78	97.77	96.46	11189	134635	145824	129840	142600	88.8	72.3	82.8
26	6	11840	6470	5305	172.6	49.2	19.02	4.32	27.48	98.18	96.48	14090	161983	176073	156300	172900	88.2	73.1	84.9
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20

TABLE 5 SERIES C

RESULTS									GUARANTEE & CALCULATED WATER RATES			CIRC. WATER		WATER RATE, UNIT,	REMARKS	FORMULAE & NOTES
TEST	LOAD	WATER RATES (Dry Steam)														
		TURBINE ACTUAL	ENGINE ACTUAL	UNIT ACTUAL		UNIT Moist Corr.	UNIT Total Corr.		TURBINE Constant Nozzle Pr.	ENGINE Constant Nozzle Pr.	UNIT Constant Nozzle Pr.	RATIO	TOTAL	EQUIVALENT I.H.P.		
	K.W.	Lbs per K.W.H.	Do	Do		Do	Do.		Do	Do.	Do.	C.W. / Dry St.	Gals per Minute	Lbs. per I.H.P.Hr. C.N.P.		
5	10220	36.98	22.48	14.57		14.0	13.53		28.42	26.03	14.42	89.0	26740	8.85	Varying Nozzle Pressure	27 = 17 / 4 + [1+(1−12)]5
6	11320	32.34	25.30	14.45		14.1	13.94		27.30	26.04	14.07	80.5	26920	8.68	Auxiliary Steam Included C.N.P.	
7	11150	30.42	26.05	14.49		13.85	13.90		27.22	26.07	14.10	78.3	25800	9.10	Constant Nozzle Pressure	28 = 27−(285−10)
8	10970	29.58	25.51	14.37		13.64	13.70		27.31	26.07	14.15	86.5	27960	8.76	Do	
9	11250	32.42	25.20	14.69		14.23	13.55		27.42	26.04	14.09	79.0	25980	9.03	Do.	
10	12440	29.89	26.01	14.76		13.95	13.36		27.00	26.08	13.88	93.5	30560	8.27	V.N.P.	
11	8990	32.91	25.79	14.89		14.36	13.78		28.29	26.99	15.10	94.5	25820	8.76	C.N.P.	
12	13240	29.15	25.71	14.72		13.90	13.39		26.90	26.46	13.90	76.5	30320	8.25	Do	
13	10240	33.76	25.52	15.13		14.51	13.99		27.83	26.10	14.41	96.0	30200	9.01	Do.	
14	11480	29.78	28.30	15.15		14.47	14.26		27.02	26.10	14.03	78.6	25000	9.14	V.N.P.	
15	11504	29.06	—	—		—	—		26.97	26.24	14.01	69.	23060	—	Aux. Included C.N.P.	
16	11526	30.22	27.42	15.02		14.64	13.56		27.10	26.12	14.01	82.	27400	8.93	C.N.P.	
17	11528	30.90	29.70	15.68		15.20	14.32		26.96	26.20	14.01	81.	29020	9.53	Aux. Included V.N.P.	
18	10740	29.26	26.84	14.78		14.41	13.56		27.31	26.30	14.22	98.	29230	9.06	C.N.P.	
19	14540	28.60	27.05	14.62		14.30	13.58		26.63	26.69	13.80	66.	26560	9.08	Do.	
20	14365	27.23	28.15	14.45		14.25	13.50		26.60	26.31	13.80	51.	20050	9.05	V.N.P.	
21	10320	27.97	26.11	14.25		13.70	13.23		27.49	26.50	14.39	85.	24520	8.52	C.N.P.	
23	13410	32.05	27.70	15.41		15.25	14.47		26.81	26.37	13.80	75.5	29520	9.81	Do.	
24	12927	28.40	26.14	14.22		14.0	14.29		26.82	26.16	13.81	78.5	27180	8.93	Do	
25	9730	28.81	27.73	14.65		14.53	13.81		27.57	27.07	14.67	101.	27200	9.33	Do	
26	11840	29.45	26.71	14.60		14.44	13.42		27.04	26.09	13.96	91.	24500	9.06	Do.	
21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37

TABLE 6 SERIES C

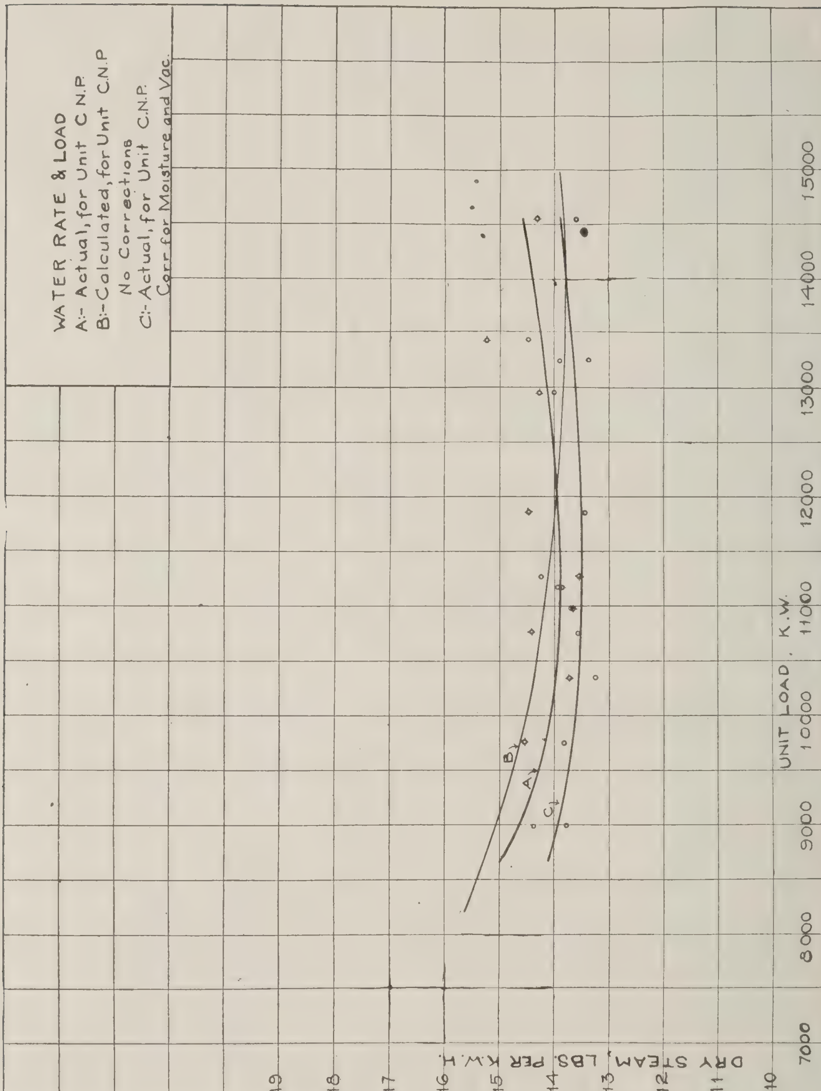


FIG. 11 SERIES C

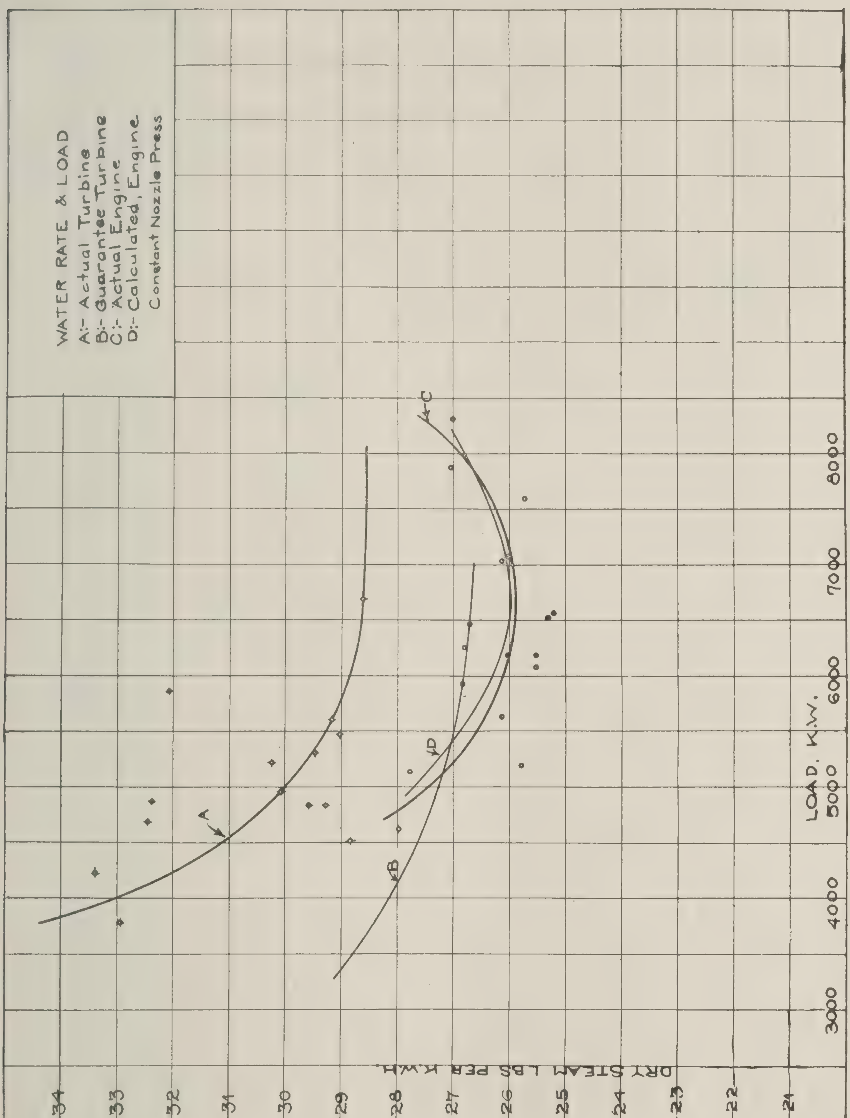


FIG. 11a SERIES C

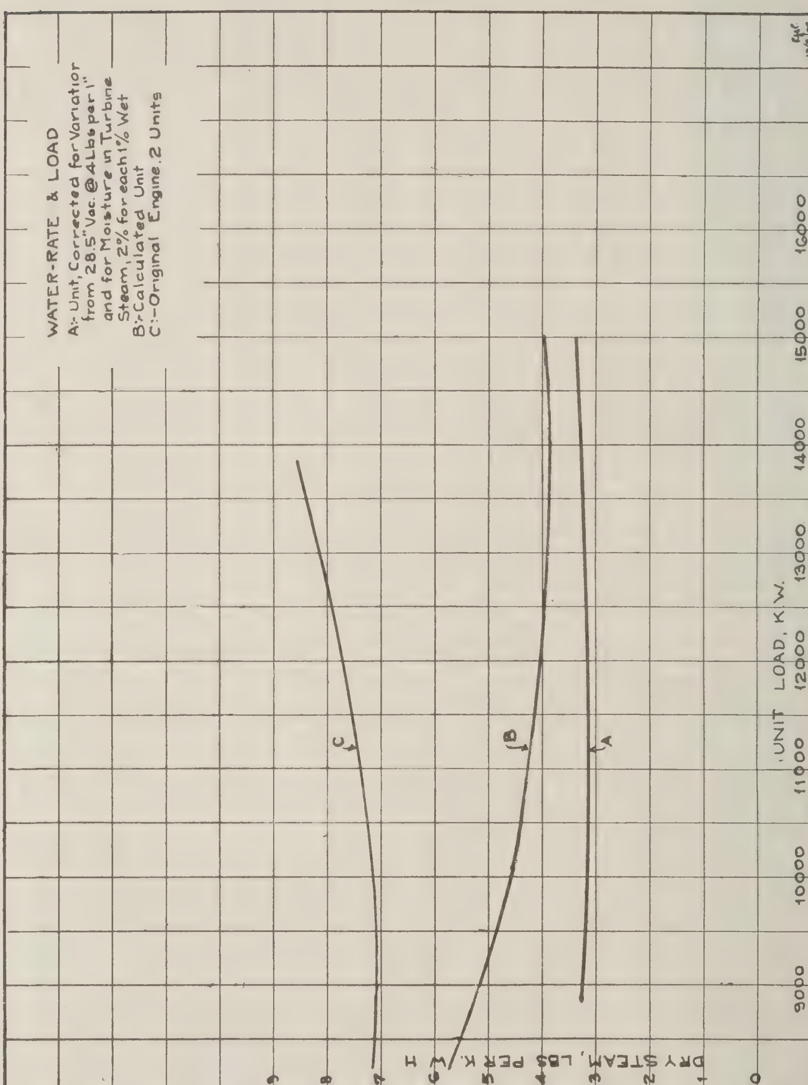


FIG. 11b SERIES C

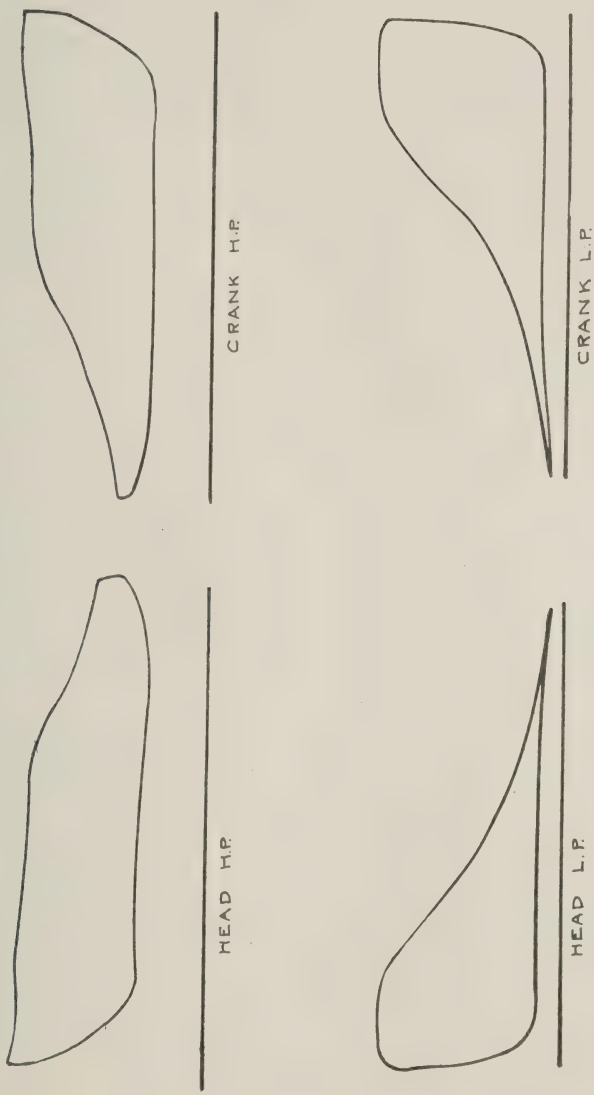


FIG. 12 SERIES C

TEST	DURATION Hrs	LOADS UNIT KW	LOADS ENG. KW	LOADS TURB. KW	PRESSURES H.P. STM Lbs/□" Absolute	PRESSURES RECEIVERS	PRESSURES TURB STM. Do	PRESSURES VACUUM Do	PRESSURES VAC 29.92" Barom "Hg	QUALITIES H.P. STM %	QUALITIES TURB STM %	WATER Lbs/Hr TOTAL CONDENSATION	WATER Lbs/Hr TURB. WATER	WATER Lbs/Hr TOTAL WATER UNIT	DRY STEAM TURBINE	DRY STEAM ENG.& UNIT	TEMPERATURES COND. HOT WELL °F	TEMPERATURES CIRC W. INJECTION °F	TEMPERATURES CIRC W DISCH. °F
27	6	9567	4875	4683	185.1	64.7	18.14	.70	28.50	99.0	93.4	8636	133840	142476	125000	141100	78.6	38.80	58.60
28	6	10527	5819	4714	189.8	64.9	16.93	.87	28.15	99.6	95.2	8570	146730	155300	139700	154700	80.4	39.53	67.04
29	2	11400	6410	5000	184.7	65.2	17.62	1.05	27.87	98.9	97.3	9601	162475	172076	158100	170300	81.3	37.40	67.77
30	4	11365	6590	4810	185.6	64.9	17.62	1.15	27.58	99.0	97.8	10088	161351	171439	157800	169700	84.5	39.20	73.90
31	6	12300	6930	5373	192.0	65.5	17.59	.90	28.12	99.3	97.8	11006	166826	177832	163200	176800	78.64	36.54	66.00
32	6	13160	7490	5717	193.4	66.3	17.66	95	27.98	99.5	97.2	12142	179776	191918	174700	191000	75.34	35.70	69.72
33	4	16085	8505	7670	192.3	65.2	20.00	1.12	27.64	99.8	94.7	7312	247090	254402	234000	254000	86.44	35.05	73.41
34	6	14223	8183	6060	192.3	65.8	17.56	1.11	27.66	99.4	95.7	9361	206261	215622	197500	214300	89.06	34.88	74.30
35	4	13295	7783	5524	194.7	65.8	17.13	1.20	27.49	99.5	95.4	11145	187214	198359	178600	197500	90.85	35.22	76.39
36	5	12217	7169	5033	192.7	64.3	16.28	1.05	27.78	99.8	93.8	13017	165908	178925	155500	178700	89.42	34.66	70.76
37	6	10043	5813	4184	190.4	63.9	16.40	1.02	27.84	99.4	93.7	9823	140932	150755	132000	149800	91.80	36.67	70.72

TEST	ACTUAL WATER-RATES ENG.	ACTUAL WATER-RATES TURB	ACTUAL WATER-RATES UNIT	CORRECTED UNIT MOISTURE	CORRECTED UNIT MOIST- & VAC	I.H.P. ENGINE H.P.	I.H.P. ENGINE L.P.	I.H.P. ENGINE TOTAL	FACTOR IHP/KW
27	28.94	27.65	14.75	14.30	14.30	3861	3502	7363	1.511
28	26.69	30.15	14.68	14.37	14.02	4138	4359	8497	1.461
29	26.57	31.62	14.94	14.74	14.11	4445	4712	9157	1.429
30	25.74	32.81	14.93	14.74	18.82	4477	4727	9204	1.397
31	25.49	30.38	14.36	14.22	13.74	4753	5045	9798	1.414
32	25.50	30.57	14.51	14.29	13.77	5092	5583	10675	1.425
33	29.86	30.80	15.79	15.32	13.46	5902	6229	12131	1.426
34	26.18	32.60	15.07	14.78	13.94	5585	6018	11603	1.417
35	25.38	32.34	14.85	14.56	13.55	5246	5682	10928	1.404
36	24.92	30.91	14.62	14.29	13.57	4873	5421	10294	1.435
37	25.77	31.54	14.93	14.60	13.94	4248	4312	8560	1.473

TABLE 7 SERIES D

NO. TEST	DATE 1910	DURATION Hrs.	LOADS TOTAL UNIT K.W.	ENGINE K.W.	TURBINE K.W.	PRESSURES MN. STEAM Abs. Lbs/□"	MN. STM. Gge. Lbs/□"	RECEIVERS Abs. Lbs/□"	REC. Gge. Lbs/□"	L.P. SEPARATOR Abs. Lbs/□"	L.P. SEP. Gge. Lbs/□"	L.P. SEP. U Tube "Hg.	VACUUM Hg. Col. "Hg.	VAC. Absolute Manom. "Hg.Ab.	VAC. Abs. Lbs.Abs	BAROMETER "Hg.	STD VAC. 29.92" Bar. "Hg.	C.W.PUMP SUCTION "Hg	C.W.PUMP DISCHARGE Gge Lbs/□"
38	Jan 11	5	16172	8384	7784	197.0	182.3	64.2	49.5	20.60	5.90	10.94	29.30	1.50	.74	30.63	28.42	12.7	12.0
39	11	5	13485	7798	5895	203.8	189.1	64.5	49.8	16.50	1.80	2.61	29.10	1.58	.78	30.59	28.34	12.2	12.9
40	12	5	13038	7314	5711	196.1	181.4	63.8	49.1	16.20	1.50	1.82	29.46	1.22	.60	30.61	28.72	11.0	12.8
41	12	5	12284	6938	5348	200.2	185.5	64.5	49.8	16.10	1.40	2.18	29.32	1.31	.64	30.57	28.69	12.6	12.1
42	13	5	11252	6248	4938	197.0	182.3	64.0	49.3	16.24	1.54	2.20	29.41	1.32	.65	30.65	28.60	9.9	14.3
43	13	5	10476	5824	4602	198.0	183.3	63.8	49.1	16.20	1.50	2.00	29.23	1.46	.72	30.58	28.46	12.6	12.8
44	14	5	9408	4940	4426	198.8	184.1	63.8	49.1	16.10	1.40	2.28	29.47	.93	.46	30.31	28.99	9.5	11.6
45	14	1¾	9712	5916	3709	198.2	183.5	64.3	49.6	10.50		-7.48	28.96	1.22	.60	30.21	28.70	11.6	11.6
46	15	½	12700	7180	5640	198.5	183.8	62.6	47.9	12.96		-2.78	29.45	1.03	.51	30.32	28.89	13.75	9.5
47	15	½	11940	7060	4780	195.3	180.6	62.6	47.9	12.35		-1.80	29.72	1.03	.51	30.32	28.89	12.9	9.9
48	15	3	9306	5865	3323	196.3	181.6	63.8 46.3	49.1 31.6	9.65		-10.49	29.13	1.20	.59	30.33	28.72	12.0	10.6
49	15	1	10940	6640	4300	194.9	180.2	49.3	35.6	11.65		-6.62	29.19	1.13	.56	30.32	28.79	12.1	10.9
50	15	4	15498	8169	7260	192.0	177.3	59.0	44.3	17.80	3.10	3.54	29.23	1.13	.56	30.32	28.79	14.6	10.4
51	17	3	11240	6753	4376	194.0	179.3	55.0	40.3	11.50		-6.71	29.25	1.19	.58	30.38	28.73	11.5	9.9
52	17	3	7200	4743	2400	196.5	181.8	41.5	26.8	7.97		-14.92	29.07	1.29	.63	30.29	28.63	10.6	10.7
53	17	3	11927	7070	4834	199.0	184.3	52.9	38.2	12.75		4.58	29.10	1.25	.61	30.26	28.67	13.2	10.1
54	26	3	14173	7820	6283	193.4	178.6	55.7	40.9	15.18	.48	1.04	29.31	.93	.46	30.03	28.99	10.1	9.5
55	26	3	8347	5403	2910	197.4	182.7	43.0	28.3	8.21		-13.48	28.90	1.15	.57	29.97	28.77	12.4	9.0
56	26	3	13033	7457	5550	197.2	182.5	54.3	39.6	14.09		-1.75	29.01	1.01	.50	29.90	28.91	13.2	9.4
57	27	3	14580	7960	6583	199.7	179.2	59.6	45.1	15.57	.87	+3.04	28.85	.85	.42	29.49	29.07	10.3	8.8
58	27	3	6673	4420	2213	197.7	183.2	40.1	25.6	7.08		-15.14	28.48	.98	.48	29.48	28.94	12.6	8.6
59	27	3	10007	6194	3804	199.1	184.6	47.3	32.0	10.35		-8.59	28.55	.94	.46	29.50	28.98	14.3	8.7
60	28	3	11820	6923	4860	195.1	180.5	51.6	36.9	12.10		-4.98	28.96	.87	.43	29.79	29.05	10.7	10.4
61	28	3	11480	6587	4893	196.8	182.2	51.4	36.7	12.34		-4.46	28.88	1.00	.49	29.79	28.92	12.4	10.2
62	28	3	15860	8440	7410	195.0	180.4	63.4	48.8	17.84	3.14	7.60	28.73	1.19	.58	29.79	28.73	13.9	9.9

TABLE 8 SERIES E AND F

NO. TEST	LOAD	H.P. THROTTLING CAL RIM. AV. MN STM. TEMP.	CAL. DISCH. TEMP.	CAL. DISCH. PRESS. Gauge	QUAL.	#1 SEPARATING. MOISTURE	FLOW	QUAL.	#2 SEPARATING MOISTURE	FLOW	QUAL.	#1 THROTTLING STM. TEMP.	DISCH. TEMP.	DISCH. PRESS. Vacuum	QUAL.	#2 THROTTLING DISCH. TEMP.	DISCH. PRESS. Vacuum	QUAL.
	KW	°F	°F	"Hg.	%	Lbs.	Lbs.	%	Lbs.	Lbs.	%	°F	°F	"Hg.	%	°F	"Hg	%
38	16172	380.6	312.1	18.27	99.5	11.55	153.43	92.9				229.2	198.4	17.76	99.2			
39	13485	383.4	282.0	25.68	97.9	10.49	117.59	91.8				217.6	184.2	20.31	98.8			
40	13038	380.2	302.9	21.81	99.2	10.10	119.41	92.2				216.7	182.0	20.74	98.8			
41	12284	381.9	297.5	19.80	98.9	9.88	119.93	92.4				216.3	180.3	20.55	98.7			
42	11252	380.6	294.8	20.77	98.7	9.73	120.16	92.6	1.42	18.77	93.0	216.7	181.5	20.60	98.7	164.0	26.12	98.1
43	10476	381.0	291.8	20.42	98.5	8.86	111.94	92.6	3.18	46.47	93.6	216.7	182.9	21.50	98.8	168.9	26.23	98.4
44	9408	381.3	297.9	21.78	98.7	8.05	99.50	92.5	1.08	18.90	94.6	216.4	179.3	21.55	98.6	177.0	25.71	98.5
45	9712	381.1	294.8	21.92	98.7	1.10	14.10	92.8	.79	11.95	93.8	195.4	172.5	23.70	99.2	159.5	27.37	98.6
46	12700	381.2	296.7	21.40	98.8	.43	10.60	96.1	.16	4.00	96.2	210.2	170.3	21.30	98.4	163.9	26.40	98.3
47	11940	379.9	304.0	20.63	99.1	.29	9.10	96.9	.16	3.60	95.8	203.3	176.0	22.48	98.9	165.7	26.72	98.6
48	9306	380.3	290.5	21.18	98.6	2.43	42.25	94.6	1.22	16.88	93.3	191.8	171.2	24.29	99.3	156.9	27.42	98.6
49	10940	379.7	299.0	21.05	98.9	.74	16.77	95.7	.41	6.90	94.4	200.3	178.3	22.62	99.3	167.3	26.76	98.7
50	15498	378.5	305.6	19.77	99.4	2.84	70.70	96.2	1.27	41.56	97.0	221.4	189.4	19.89	98.8	182.9	25.40	98.6
51	11240	379.4	292.6	21.93	98.6	2.04	51.67	96.2	1.15	19.96	94.5	199.8	174.8	22.68	99.0	165.0	27.04	99.1
52	7200	380.4	288.8	20.71	98.4	2.04	36.77	94.7	.93	14.05	93.7	182.4	163.0	25.21	99.3	148.0	27.75	98.6
53	11927	381.4	294.4	20.64	98.6	1.79	56.20	95.3	1.11	21.66	95.1	204.7	177.0	21.96	99.0	168.8	26.73	98.7
54	14173	379.1	296.9	22.41	98.9	2.13	56.70	96.3				213.3	191.4	19.92	99.4			
55	8347	380.8	292.9	20.87	98.6	1.57	38.91	96.1				184.2	165.2	24.20	99.3			
56	13033	380.7	298.2	20.25	98.7	1.63	61.64	97.4				209.7	184.9	20.81	99.1			
57	14580	379.2	298.8	21.75	99.0	2.06	70.87	97.2				215.3	188.9	18.77	99.0			
58	6673	380.9	298.4	21.80	99.0	1.79	33.30	94.9				178.4	161.1	24.41	99.4			
59	10007	381.5	287.0	21.53	98.2	2.62	47.12	94.7				194.8	171.1	22.70	99.2			
60	11820	379.8	299.9	21.34	99.0	1.56	54.77	97.2				202.4	175.0	21.67	99.0			
61	11480	380.5	302.8	21.00	99.1	1.55	55.79	97.3				203.3	178.4	21.57	99.0			
62	15860	379.7	299.7	20.63	99.0	2.19	80.02	97.3				221.9	197.6	17.87	99.3			

TABLE 9 SERIES E AND F

NO. TEST	THOMAS CALORIMETER: WATT HRS. W.H	THOMAS CALORIMETER: STM. TEMP. °F	THOMAS CALORIMETER: CAL DISCH. TEMP °F	THOMAS CALORIMETER: CAL DISCH. WATER Lbs	THOMAS CALORIMETER: QUAL. %	#1 COMB. QUAL. %	#2 COMB. QUAL. %	AV'GE QUAL. %	EXHAUST FROM ENGINE: SEPARATING CAL.: MOISTURE Lbs.	SEPARATING CAL.: FLOW Lbs.	SEPARATING CAL.: QUAL. %	THROTTLING CAL.: STM. TEMP °F	THROTTLING CAL.: CAL. DISCH. TEMP. °F	THROTTLING CAL.: CAL. DISCH. PRESS. Vac "Hg	THROTTLING CAL.: QUAL. %	COMB QUAL. %	QUAL. by Separator Water x Turb Stm. Qual. %
38	6600	229.2	414.6	93.90	84.1	92.2		90.2									
39	5623	217.6	430.6	84.10	86.3	90.7		89.4									
40	4758	216.7	429.9	93.60	92.4	91.1		91.5									
41	3242	216.3	412.7	70.20	93.2	91.2		91.9									
42	4268	216.7	426.8	110.20	94.4	91.4	91.2	93.0									
43	5000	216.7	408.3	116.50	94.3	91.5	92.1	92.6									
44	4575	216.4	424.6	88.80	91.9	91.2	93.2	92.1									
45	1519	195.4	445.2	30.90	94.9	92.0	92.5	93.1									
46	500	210.2	356.9	20.40	98.5	94.6	94.6	95.9									
47	350	203.3	454.2	27.40	95.6	95.8	94.5	95.3									
48	2100	191.8	420.0	37.92	91.7	93.9	92.0	92.5									
49						95.0	93.2	94.1									
50	2377	221.4	346.5	81.70	95.8	95.0	94.4	95.1									
51	2100	199.8	420.7	47.70	95.4	95.2	93.6	94.7									
52	2300	182.4	395.5	58.40	96.5	94.0	92.4	94.3									
53	2500	204.7	433.2	59.00	96.2	94.3	93.8	94.8									
54	650	213.3	276.7	44.05	98.0	95.7		96.9	2.53	26.90	91.4	213.8	180.5	23.29	98.7	90.2	89.6
55	600	184.2	293.6	28.10	97.4	95.4		96.4	2.05	15.17	88.1	185.6	148.7	26.71	98.5	86.8	90.2
56	600	209.7	268.5	44.40	98.1	96.5		97.3	2.53	21.70	89.5	209.5	167.7	25.41	98.6	88.2	90.3
57	600	215.3	260.8	56.10	98.4	96.2		97.3	1.92	21.40	91.8	216.2	172.1	25.16	98.4	90.3	89.3
58	450	178.4	267.7	24.90	97.0	94.4		95.7	1.88	17.12	90.7	177.9	148.3	25.80	98.8	89.0	89.7
59	600	194.8	265.5	32.80	97.2	94.0		95.6	2.28	15.72	87.3	195.3	154.0	26.02	98.5	86.0	89.8
60	600	202.4	274.2	39.90	98.2	96.3		97.3	2.49	19.17	88.6	203.4	163.4	25.76	98.5	87.3	90.3
61	600	203.3	272.8	41.40	98.3	96.3		97.3	2.34	16.95	87.9	204.1	160.1	26.07	98.3	86.4	91.2
62	700	221.9	274.3	55.40	98.1	96.6		97.4	2.11	24.65	92.1	222.8	178.4	24.73	98.5	90.7	89.8

TABLE 10 SERIES E AND F

TEST	WEST MN STM. °F	EAST MN STM °F	AVGE MN. STM °F	WEST RECEIVER °F	EAST REC °F	AVGE. REC. °F	EAST ENG. EXH. °F	EAST SEPARATOR INLET °F	SEPARATOR OUTLET °F	TURBINE BOWL °F	CONDENSER °F	HOT WELL WATER °F	CIRC. WATER INJECTION °F	CIRC. WATER DISCHARGE °F	
38	381.0	380.2	380.6	297.0	297.4	297.2	230.5	230.1	229.2	228.5	90.90	71.20	37.66	57.28	
39	384.1	382.8	383.4	297.3	297.4	297.4	218.7		217.6	217.0	94.60	76.23	39.85	62.51	
40	380.3	380.1	380.2	296.1	297.5	296.8	218.0	217.6	216.7	216.1	84.45	70.97	32.03	51.84	
41	382.4	381.4	381.9	297.0	297.7	297.3	217.9	217.2	216.3	216.3	87.17	73.41	33.95	56.76	
42	380.2	380.9	380.6	296.9	297.0	297.0	218.6	218.6	216.9	216.9	87.18	75.25	31.82	56.67	
43	380.6	381.3	381.0	296.8	296.6	296.7	217.9		216.7	216.3	89.32	80.17	32.75	57.14	
44	380.3	382.2	381.3	296.9	296.5	296.7	217.6		216.4	216.1	73.87	59.42	31.63	41.98	
45	380.3	381.8	381.1	296.8	297.7	297.3	208.2		195.4	196.7	86.72	66.20	32.22	43.97	
46	379.7	382.7	381.2	295.0	296.0	295.5	209.0		210.2	208.0	77.20	63.70	40.20	53.10	
47	379.0	380.8	379.9	295.3	296.0	295.6	205.0		203.3	204.0	80.00	63.00	36.00	46.85	
48	379.6	381.0	380.3	297.1 276.3	296.4 275.8	296.7 276.1	193.7 190.5		191.8	192.5 185.4	84.86	60.58	32.90	42.06	
49	377.8	381.6	379.7	279.6	280.4	280.0	202.0		200.3	201.0	83.50	62.60	31.38	53.86	
50	376.7	380.2	378.5	291.5	291.5	291.5	222.6		221.4	221.0	82.23	65.69	31.51	50.08	
51	379.0	379.8	379.4	287.2	286.8	287.0	203.4		199.8	199.8	83.16	65.18	37.50	43.10	
52	379.7	381.0	380.4	268.9	270.2	269.6	183.8		182.4	180.8	85.70	58.15	31.47	39.81	
53	380.5	382.3	381.4	285.5	283.2	284.4	205.7		204.7	204.6	83.65	65.65	31.46	45.37	
54	376.3	381.8	379.1	287.1	288.7	287.9	215.6		213.3	213.3	81.20	57.26	33.54	47.77	
55	378.0	383.5	381.8	271.8	271.7	271.7	187.0		184.2	183.8	81.00	58.64	33.08	41.30	
56	378.4	383.0	380.7	285.8	286.5	286.2	209.3		209.7	207.6	74.70	60.72	33.40	50.10	
57	377.0	381.4	379.2	293.0	291.6	292.3	217.5		214.9	215.5	67.41	53.21	33.28	47.60	
58	379.4	382.5	380.9	267.5	267.9	267.7	177.4		177.4	176.3	78.20	53.85	33.18	39.46	
59	380.0	383.0	381.5	277.5	277.4	277.5	195.7		194.8	194.1	78.30	59.12	33.40	43.66	
60	377.9	381.7	379.8	283.2	283.0	283.1	204.4		202.4	202.4	74.04	59.60	33.25	46.00	
61	378.5	382.4	380.5	283.0	282.5	282.8	205.0		203.3	203.2	77.18	61.74	33.06	47.23	
62	377.5	381.8	379.7	296.3	296.3	296.3	223.9		221.9	221.8	82.42	62.79	33.64	56.00	

TABLE 11 SERIES E AND F

NO	LOAD	TURB. WAT. WGHD	TURB WAT. VENTURI	SEP. & TRAPS WGHD	SEP WATER	TRAPS VENTURI	TOT ENG. WATER	DRY STM ENG	DRY STM. TURB.	ACT. UNIT	ACT ENG	ACT. TURB.	CORR UNIT MOIST ONLY	CORR UNIT TOTAL CORR.		AV HP.	AV L.P	TOTAL	FACTOR IHP/KW
	WATER PER HOUR									WATER RATES						I.H.P			
	K.W.	LBS PER HR	LBS PER HR	LBS PER HR	LBS PER HR	LB PER HR	LBS PER HR	LBS PER HR	LBS PER HR	LBS PER K.W.HR	LBS PER K.W.HR	LBS PER K.W.HR.	LBS PER K.W.HR	LBS PER K.W.HR.		I.H.P	I.H.P.	I.H.P	
38	16172	237480		7425	7175	250	244905	243700	214200	15.07	29.54	27.51	14.40	14.10		5640	6117	11757	1.404
39	13485	187193		9883	5881	4002	197076	193000	167400	14.32	24.76	28.38	13.48	13.10		5104	5847	10951	1.404
40	13038	170661		12271	6281	5980	182932	181500	156200	13.92	24.78	27.35	13.44	13.42		4931	5650	10581	1.447
41	12284	160229		12064	3628	8436	172293	170400	147200	13.88	24.57	27.51	13.39	13.28		4800	5306	10106	1.457
42	11252	147142		10927	2607	8320	158069	156100	136900	13.87	24.99	27.72	13.54	13.42		4250	4330	8580	1.374
43	10476	138766		9939	1801	7138	147153	145000	128500	13.85	24.90	27.91	13.47	13.21		4227	4280	8507	1.460
44	9408	122689		8387	227	8160	131076	129400	113000	13.75	26.19	25.53	13.32	13.59		3771	3490	7261	1.470
45	9712	121870		13160	4910	8250	135030	133300	113500	13.72	22.53	30.59	13.49	13.45		3819	4367	8186	1.385
46	12700	160960		18874	11242	7632	179834	177700	154400	13.99	24.75	27.38	13.62	13.77		4994	6313	11307	1.575
47	11940	142740		17385	8935	8450	160125	158700	136800	13.30	22.49	28.45	13.15	13.31		4675	5736	10411	1.475
48	9306	114372		12555	4744	7811	126927	125100	105800	13.44	21.34	31.84	13.25	13.25		4129	4321	8450	1.441
49	10940	134785		16565	8257	8308	151350	149700	126900	13.69	22.40	29.50	13.39	13.46		5039	4545	9584	1.444
50	15498	202656		19161	16661	2500	221817	220400	192900	14.22	26.99	26.55	13.96	14.03		5784	5825	11609	1.421
51	11240	133462		16689	8399	8290	150151	148100	126500	13.19	21.95	28.89	13.04	13.04		4585	5244	9829	1.456
52	7200	89158		10861	7170	3691	100019	98400	84080	13.67	20.75	35.04	13.52	13.43		3662	3478	7140	1.505
53	11927	143897		17901	8164	9737	161798	159500	136500	13.38	22.57	28.22	13.13	13.08		4995	4897	9892	1.400
54	14173	176565		19180	12930	6250	195745	193600	171050	13.66	24.75	27.22	13.54	13.81		5430	5370	10800	1.400
55	8347	99210		13733	6183	7550	112943	111400	95620	13.34	20.60	32.85	13.23	13.28		3907	3792	7699	1.425
56	13033	158112		18849	11159	7690	176961	174700	153850	13.40	23.43	27.72	13.28	13.47		5115	5120	10235	1.373
57	14580	182508		20011	14741	5270	202599	200700	177600	13.75	25.19	26.95	13.65	14.00		5376	5673	11049	1.388
58	6673	81533		11225	4900	6325	92958	91800	78000	13.76	20.78	35.27	13.65	13.87		3282	3170	6452	1.460
59	10007	120863		14578	6478	8100	135441	133050	115600	13.30	21.48	30.38	13.13	13.39		4452	4294	8746	1.413
60	11820	141930		17833	9873	7960	159763	158200	138100	13.38	22.84	28.41	13.28	13.61		5108	4796	9904	1.431
61	11480	143958		17101	8791	8310	161059	159600	141500	13.92	24.45	24.22	13.75	13.95		4728	4594	9322	1.414
62	15850	206603		19762	15705	4057	226365	224200	201300	14.14	26.58	26.56	13.99	13.99		5750	6538	12288	1.456

TABLE 12 SERIES E AND F

NO. TEST	TOT. LOAD	B.T.U. DISTRIBUTION : UNIT IS 1000000 B.T.U. SUPPLIED TO UNIT		SUPPLIED TO TURB ONLY		ENG. K.W. OUTPUT		TURB. K.W. OUTPUT		TOTAL K.W. OUTPUT		TO CONDENSER		TO HOT WELL		LOST BY RADIA. ETC.	
	KW	BTU	%	BTU	%	BTU	%	B.T.U	%	BTU	%	BTU	%	BTU	%	BTU	%
38	16172	292.0	100	252.5	86.5	28.61	9.8	26.58	9.1	55.19	18.9	216.6	74.2	9.31	3.2	10.89	3.7
39	13485	232.6		196.6	84.6	26.36	11.3	19.88	8.6	46.27	19.9	168.4	72.4	8.28	3.6	9.64	4.1
40	13038	218.0		182.5	83.7	24.97	11.5	19.49	8.9	44.46	20.4	156.4	71.7	6.65	3.1	10.53	4.8
41	12284	204.7		171.8	83.9	23.67	11.6	18.25	8.9	41.92	20.5	146.9	71.7	6.64	3.2	9.23	4.5
42	11252	187.7		159.5	85.0	21.35	11.4	16.95	9.0	38.37	20.5	136.2	72.5	6.36	3.4	6.85	3.7
43	10476	174.5		149.8	85.9	19.93	11.4	15.76	9.0	35.68	20.5	127.3	73.0	6.69	3.8	4.77	2.7
44	9408	155.8		131.9	84.6	16.87	10.8	15.12	9.7	32.09	20.6	113.4	72.7	3.36	2.2	7.03	4.5
45	9712																
46	12700																
47	11940																
48	9306	150.5		122.3	81.3	20.14	13.4	11.46	7.6	31.61	21.0	107.6	71.5	3.27	2.2	8.06	5.4
49	10940																
50	15498	264.2		224.2	84.8	27.95	10.6	24.85	9.4	52.56	19.9	192.5	72.8	6.83	2.6	12.05	4.6
51	11240	178.0		146.0	81.8	23.16	13.0	15.05	8.4	37.85	21.3	126.5	70.9	4.43	2.5	8.84	5.0
52	7200	118.4		96.41	81.4	16.25	13.7	8.26	7.0	24.37	20.6	85.8	72.5	2.33	2.0	5.74	4.8
53	11927	191.8		157.8	82.3	24.16	12.6	16.53	8.6	40.66	21.2	136.4	71.2	4.84	2.5	9.84	5.1
54	14173	232.4		197.8	83.4	26.76	11.3	21.53	9.1	48.29	20.4	171.8	72.4	4.46	1.9	7.84	3.3
55	8347	133.8		109.55	81.9	18.48	13.8	9.97	7.5	28.45	21.3	97.9	73.2	2.64	2.0	5.77	4.3
56	13033	210.0		175.5	83.6	25.46	12.1	18.97	9.0	44.43	21.2	151.9	72.3	4.54	2.2	9.04	4.3
57	14580	240.7		205.4	85.4	27.20	11.3	22.51	9.4	49.74	20.7	179.0	74.4	3.87	1.6	8.10	3.4
58	6673	110.4		89.25	80.9	15.13	13.7	7.59	6.9	22.73	20.6	79.8	72.4	1.78	1.6	6.02	5.5
59	10007	160.0		133.6	83.2	21.13	13.2	12.99	8.1	34.12	21.3	116.7	72.9	3.28	2.1	5.87	3.7
60	11820	189.8		158.9	83.7	23.67	12.5	16.62	8.8	40.29	21.2	138.4	72.9	3.92	2.1	7.23	3.8
61	11480	191.8		161.5	84.2	22.47	11.7	16.70	8.7	39.17	20.4	140.5	73.3	4.28	2.2	7.83	4.1
62	15860	269.2		232.2	86.3	28.82	10.7	25.31	9.4	54.11	20.1	200.5	74.5	6.36	2.4	8.18	3.0

TABLE 13 SERIES E AND F

NO. TEST	LOAD UNIT	RANKINE EFFICIENCIES ENG. %	TURB %	UNIT %	THERM EFFIC REF'D TO RANKINE ENG %	TURB %	UNIT %	THERMAL EFF RECOV. HW. WATER ENG %	TURB %	UNIT %	HEAT TO COND RATIO CW/ST.	B.T.U per □"/Sec.	B.T.U. per □"/Sec. per ° Rise	REMARKS
38	16172	16.7	18.2	30.1	70.3	60.0	65.0	11.7	10.9	19.5	53.5	2.58	.0594	C.N.P. Auxiliaries Exhaust to Heaters
39	13485	16.8	16.8	29.1	81.5	63.5	70.5	13.7	10.7	20.5	48.2	1.99	.0459	Do. " " "
40	13038	17.8	18.4	30.8	76.9	60.0	68.5	13.7	11.0	21.1	52.9	1.8.6	.0437	Do. " " "
41	12284	17.6	17.7	30.4	78.5	62.3	69.7	13.8	11.0	21.2	46.4	1.77	.0428	Do. " " "
42	11252	17.4	17.8	30.3	76.4	61.8	70.1	13.6	11.0	21.2	42.7	1.65	.0384	Do. " " "
43	10476	17.3	17.7	30.2	78.9	60.5	70.7	13.6	11.4	21.4	44.2	1.53	.0345	Do. " " "
44	9408	17.6	18.8	29.4	73.2	62.5	71.8	12.9	11.7	21.1	107.7	1.37	.0369	Do. " " "
48	9306	20.5	15.6	30.5	74.9	61.2	71.0	15.4	9.5	21.7	118.	1.26	.0266	V.N.P. " " "
51	11240	19.7	16.4	30.8	76.7	53.6	72.0	15.1	10.5	22.2	150	1.28	.0298	Do. " " "
54	14173	18.2	18.7	31.2	74.5	59.1	68.1	13.6	11.1	21.2	77.1	2.08	.0512	Do. " " "
55	8347	21.2	17.5	30.5	74.6	52.6	71.5	15.8	9.2	21.8	112	0.92	.0210	Do " " "
56	13033	18.2	18.4	31.0	78.5	59.4	6.90	14.2	10.9	21.7	63.4	1.87	.0566	Do. " " "
57	14580	18.0	19.5	32.2	74.0	57.0	65.4	13.6	11.1	21.0	77.5	2.16	.0800	Do. " " "
58	6673	21.3	14.6	31.4	73.2	59.0	67.0	15.6	8.6	21.0	15.3	2.55	.0608	Do. " " "
59	10007	19.5	16.7	31.3	78.5	60.0	69.8	15.3	10.0	21.8	108	1.41	.0336	Do. " " "
60	11820	19.5	20.2	32.2	74.8	52.9	67.6	14.5	10.7	21.8	82.9	1.67	.0485	Do. " " "
61	11480	19.6	17.6	31.4	69.8	60.4	66.9	13.7	10.7	21.0	108	2.40	.0885	Do. Auxiliaries exhaust into Separator
62	15860	17.2	18.7	30.0	74.0	59.4	68.9	12.7	11.1	20.7	49.0	2.52	.0670	Do. " " " Heaters.

TABLE 14 SERIES E AND F

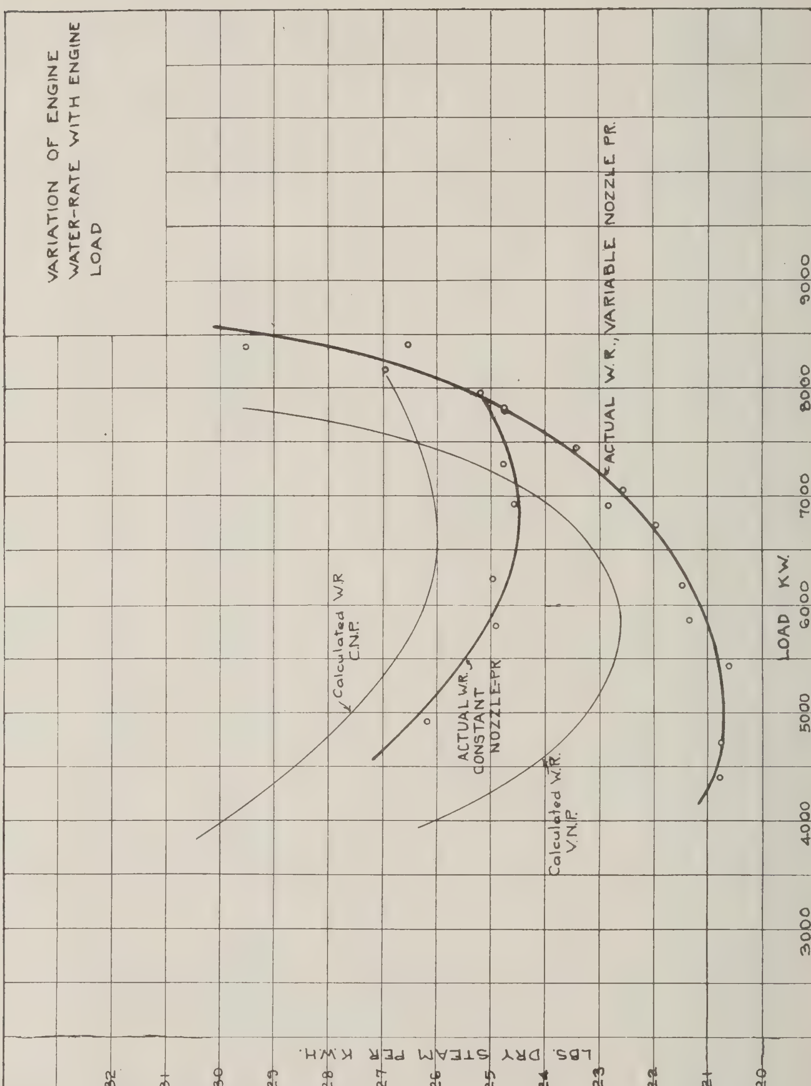


FIG. 13 SERIES E AND F

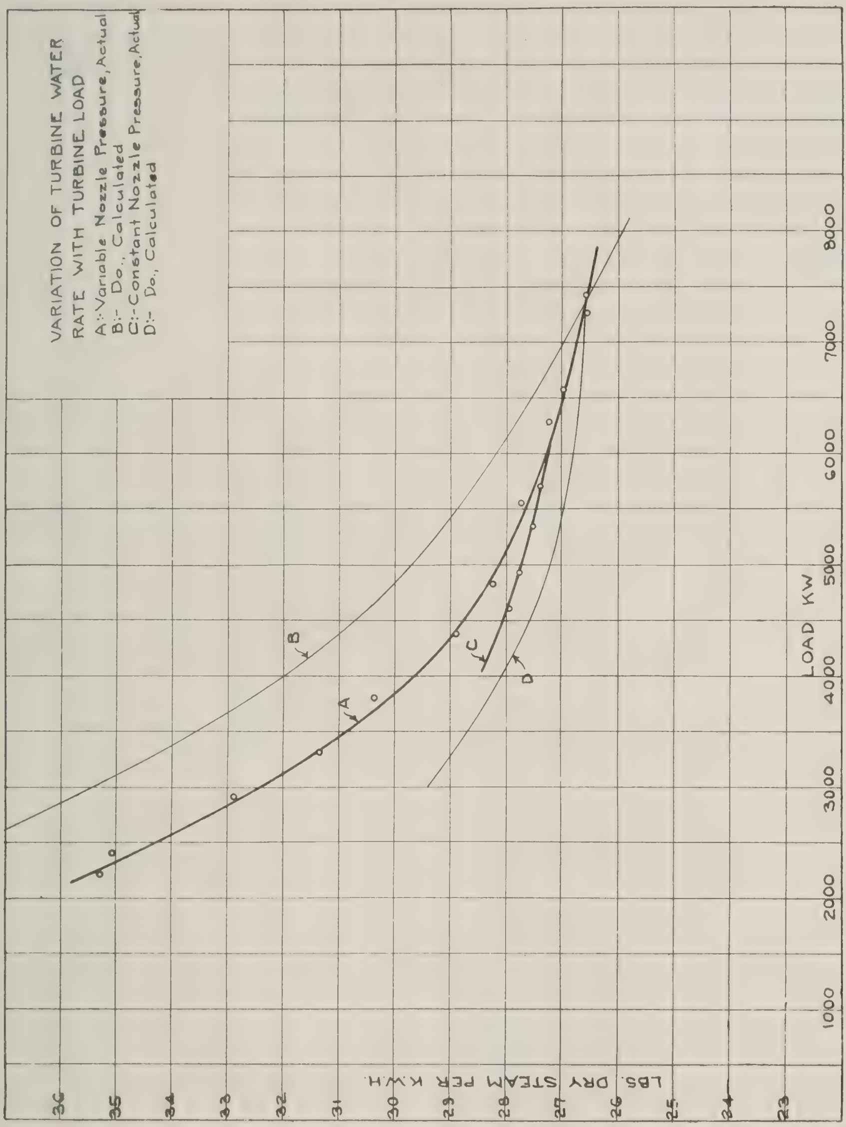


Fig. 14 Series E and F

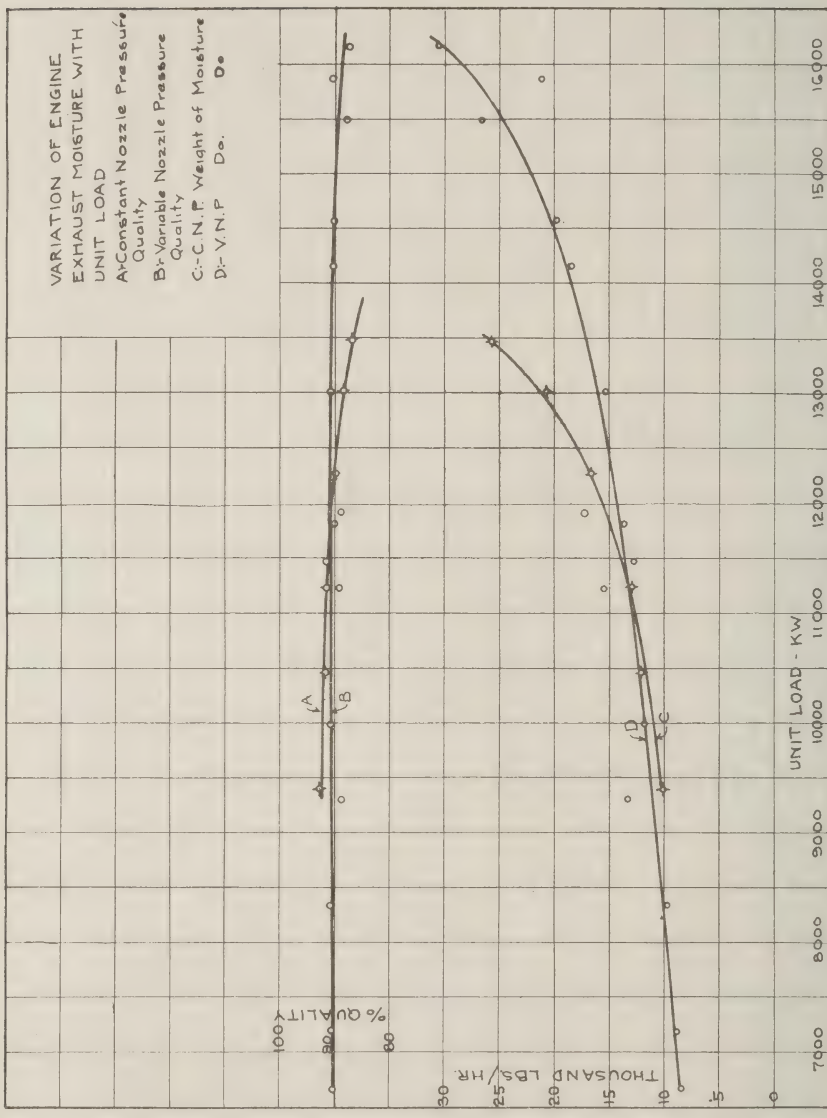


Fig. 15 Series E and F.

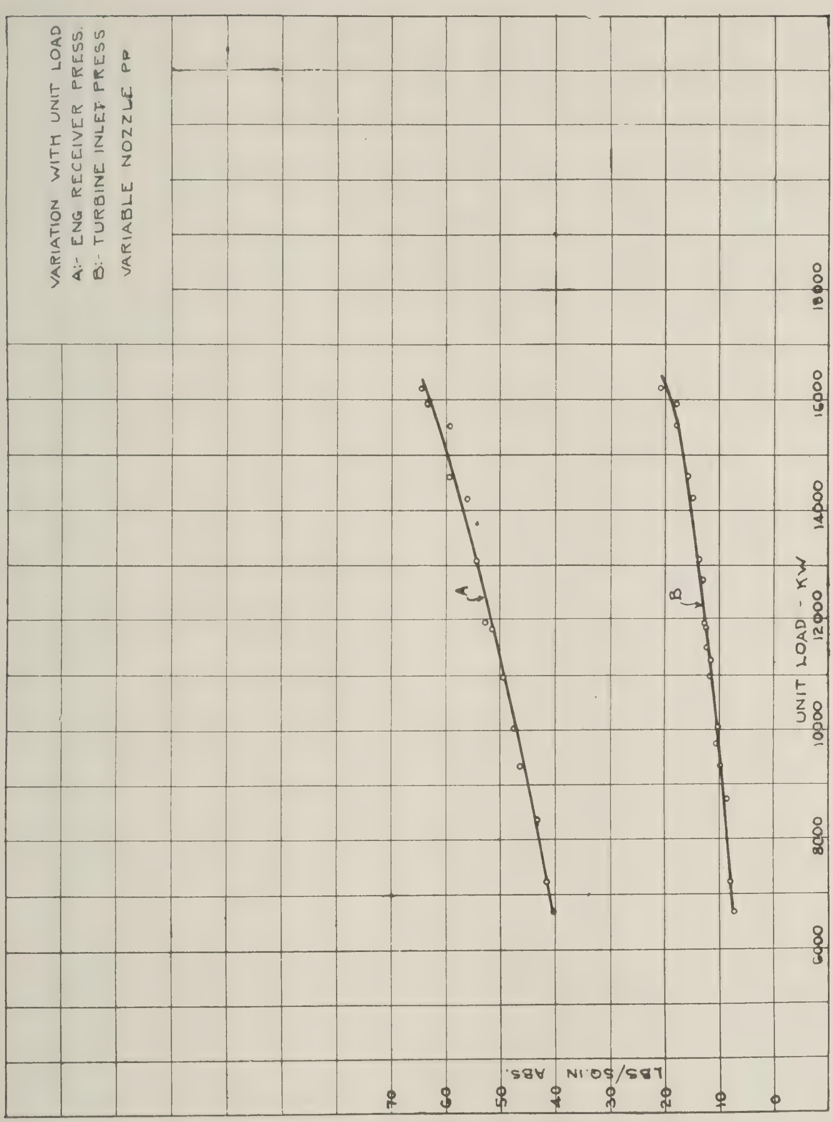


FIG. 16 SERIES E AND F

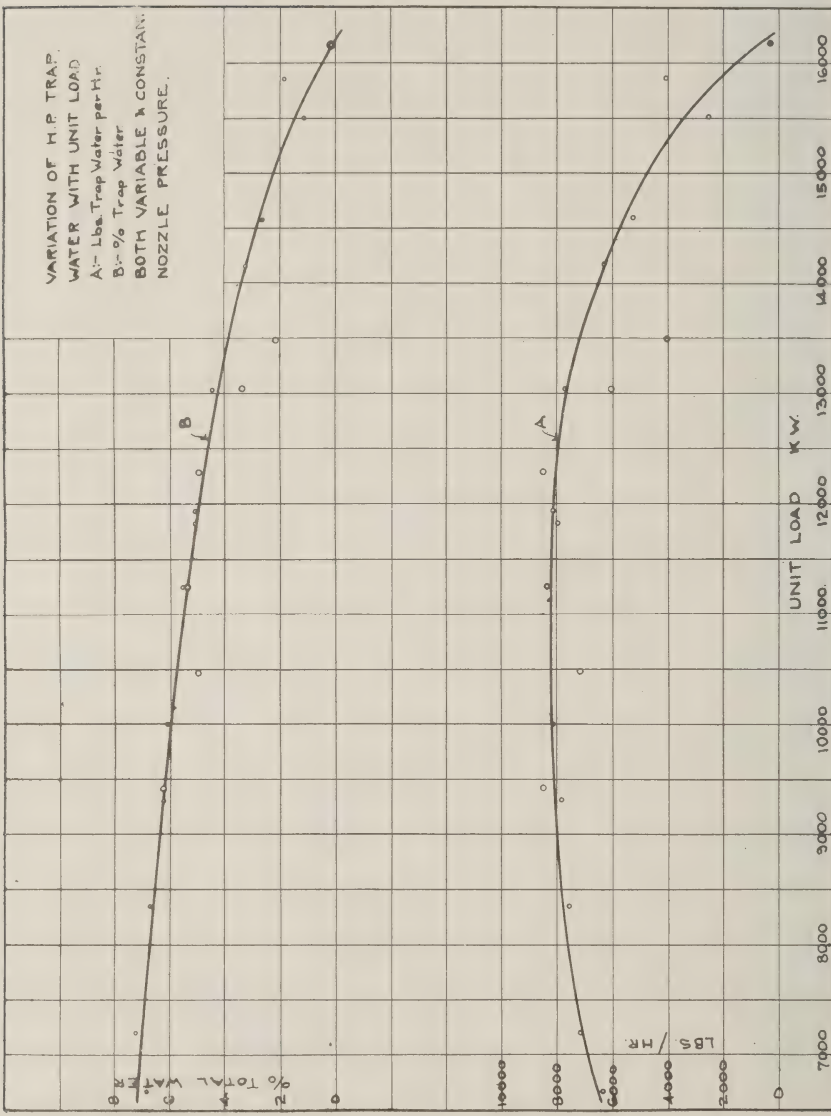


FIG. 17 SERIES E AND F

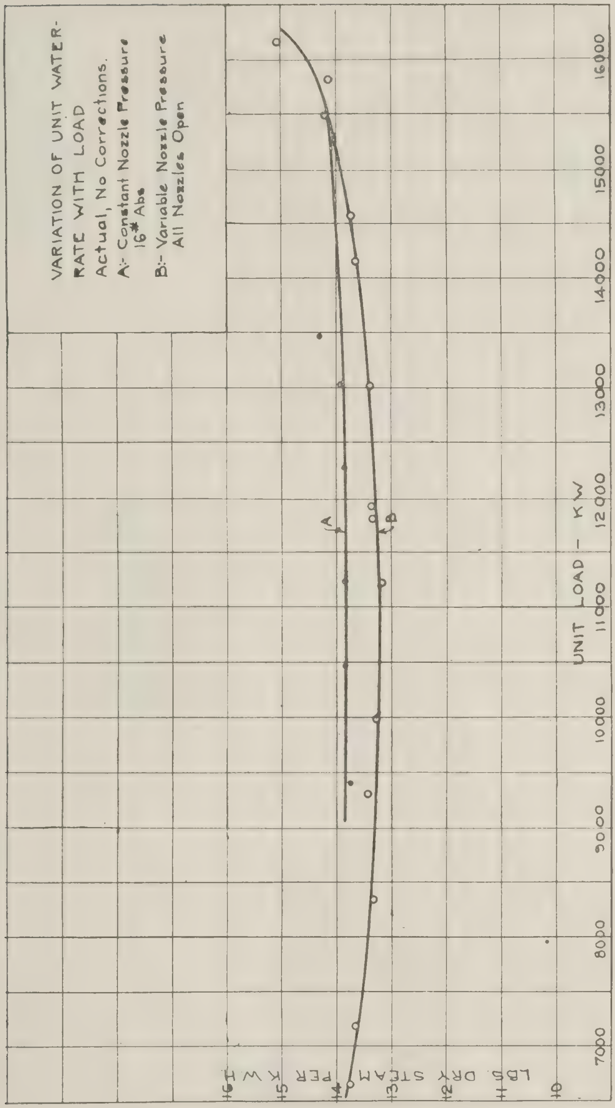


FIG. 18 SERIES E AND F

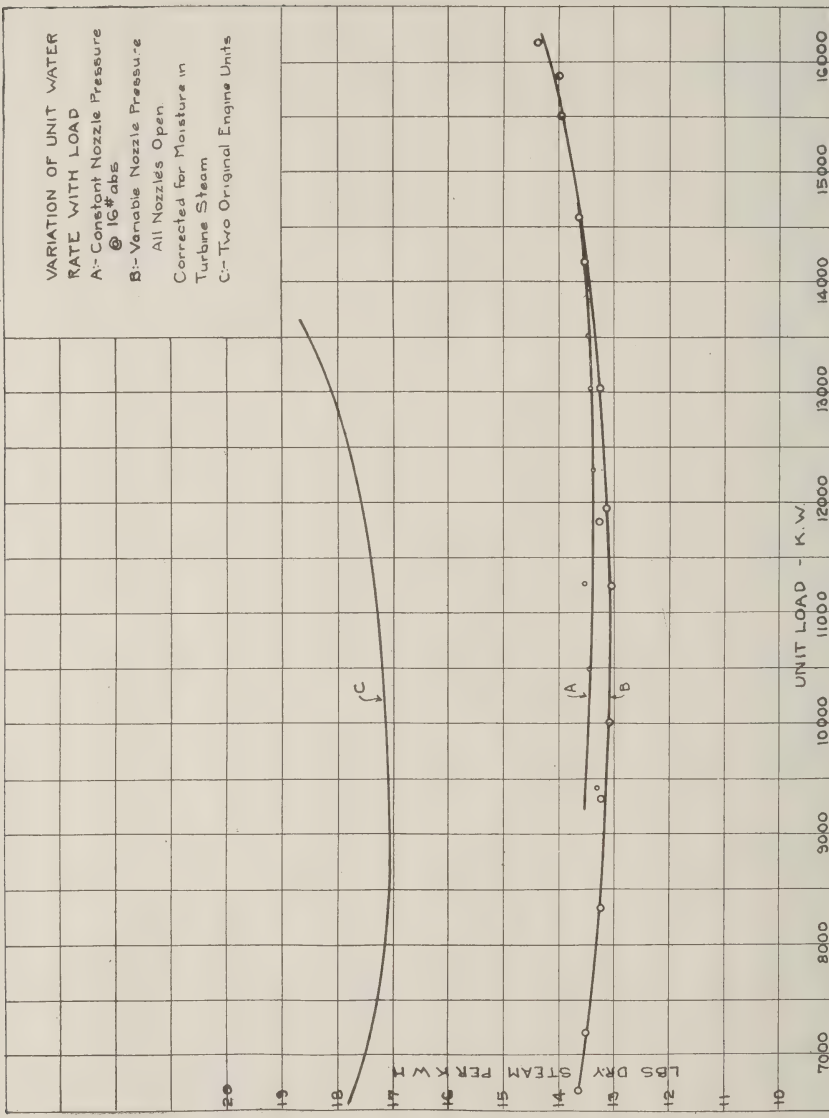


FIG. 19 SERIES E AND F

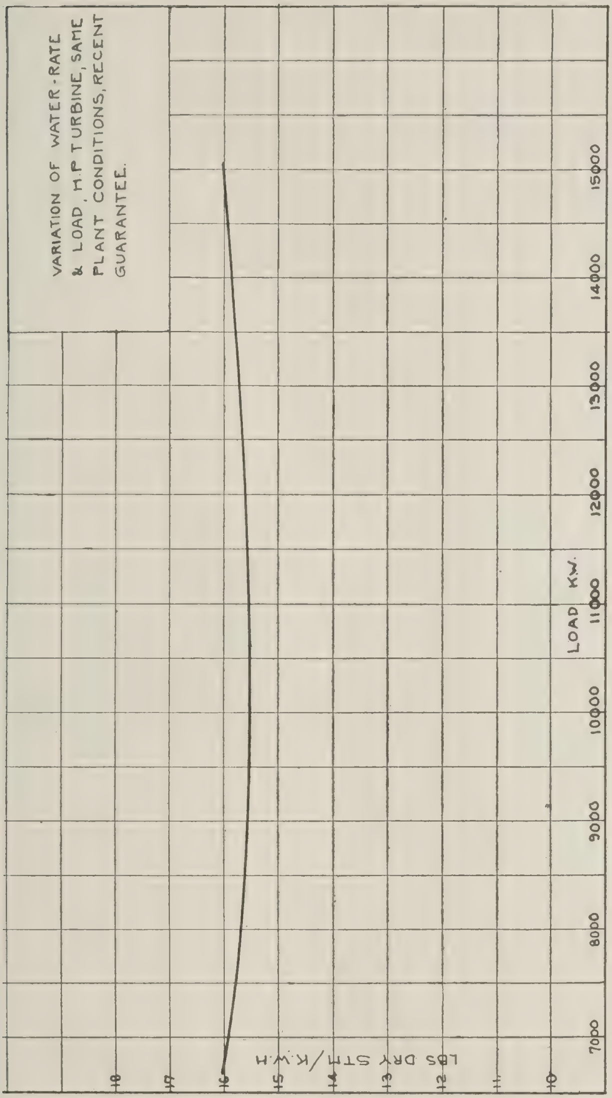


FIG. 19a SERIES E AND F

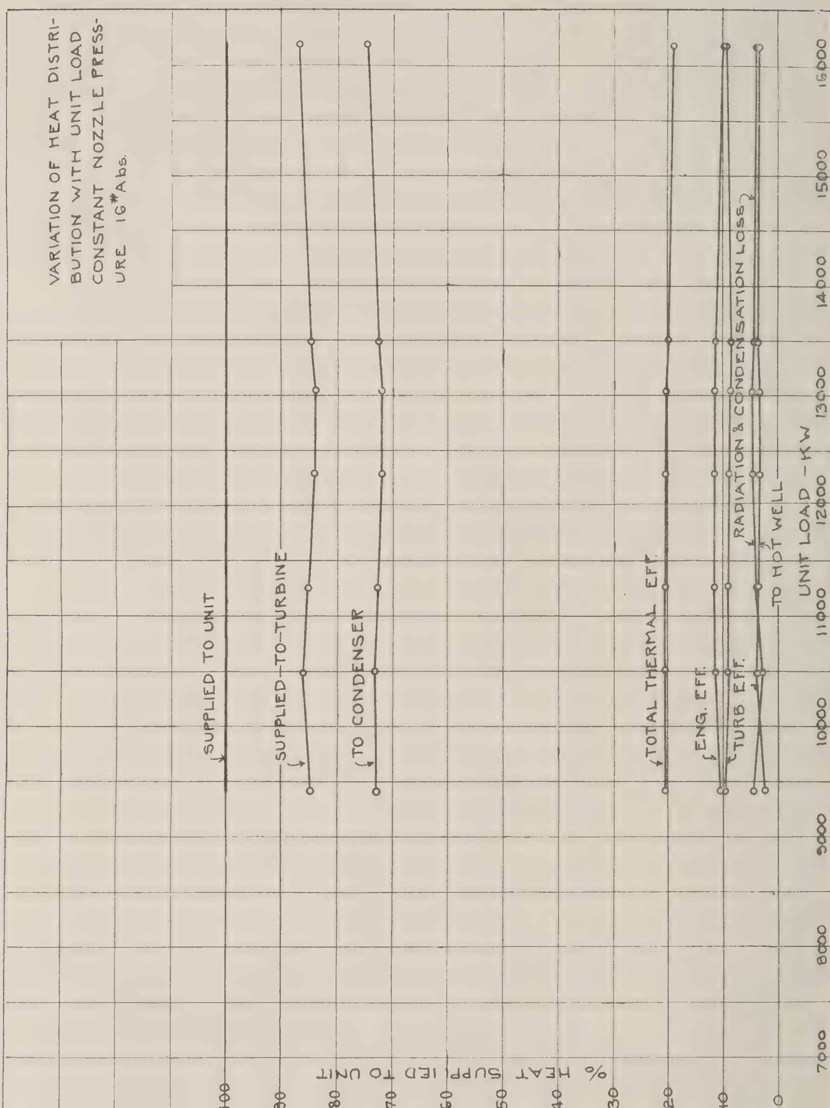


Fig. 20 Series E

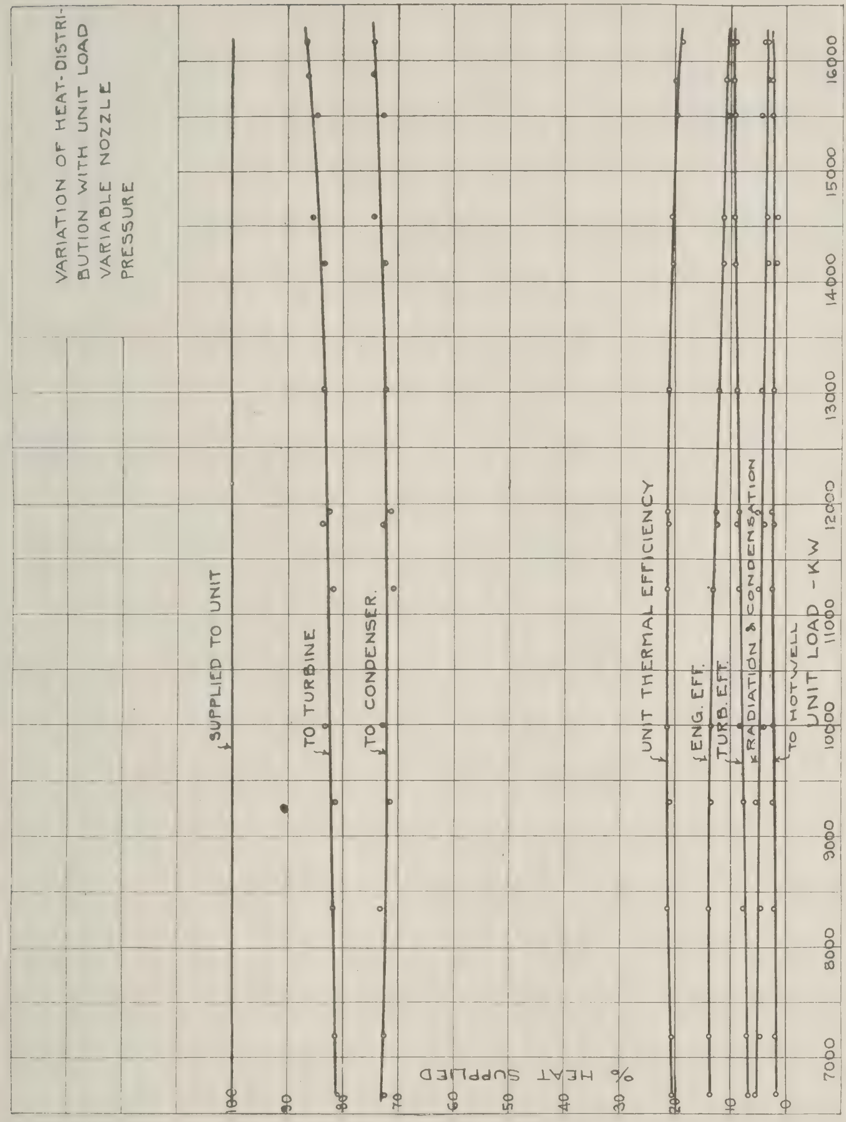


FIG. 21 SERIES F

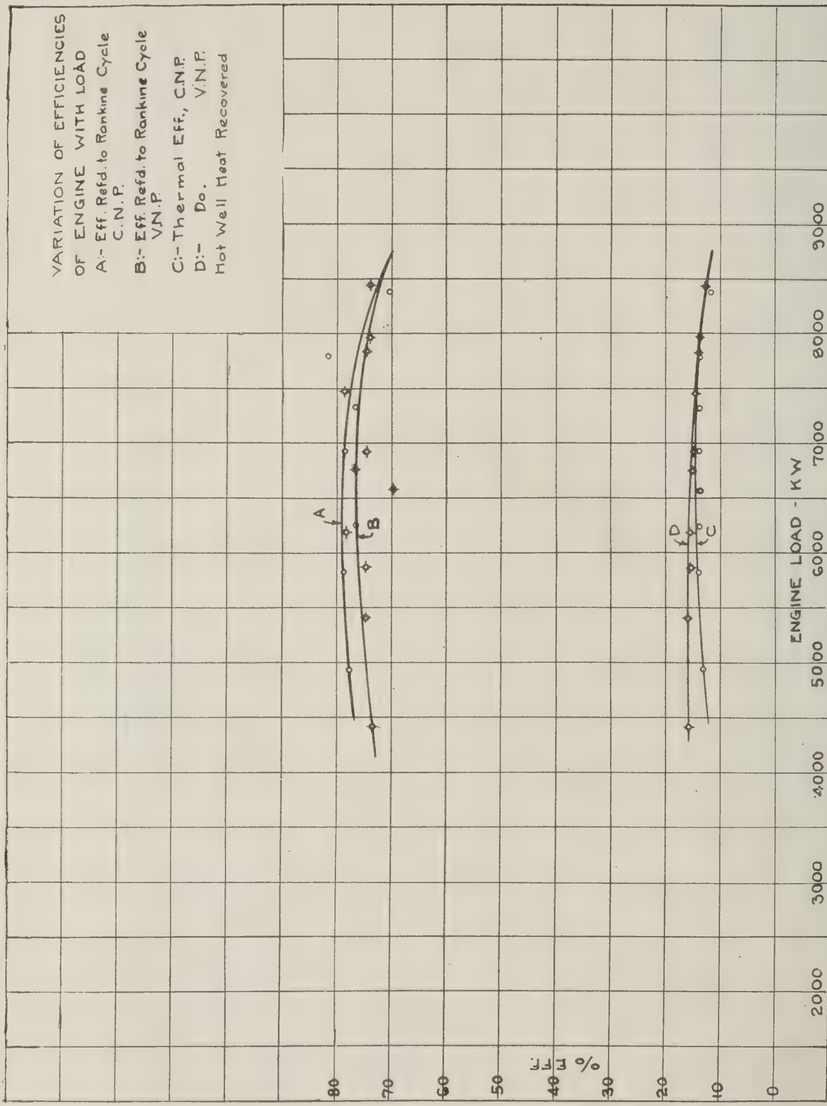


FIG. 22 SERIES E AND F

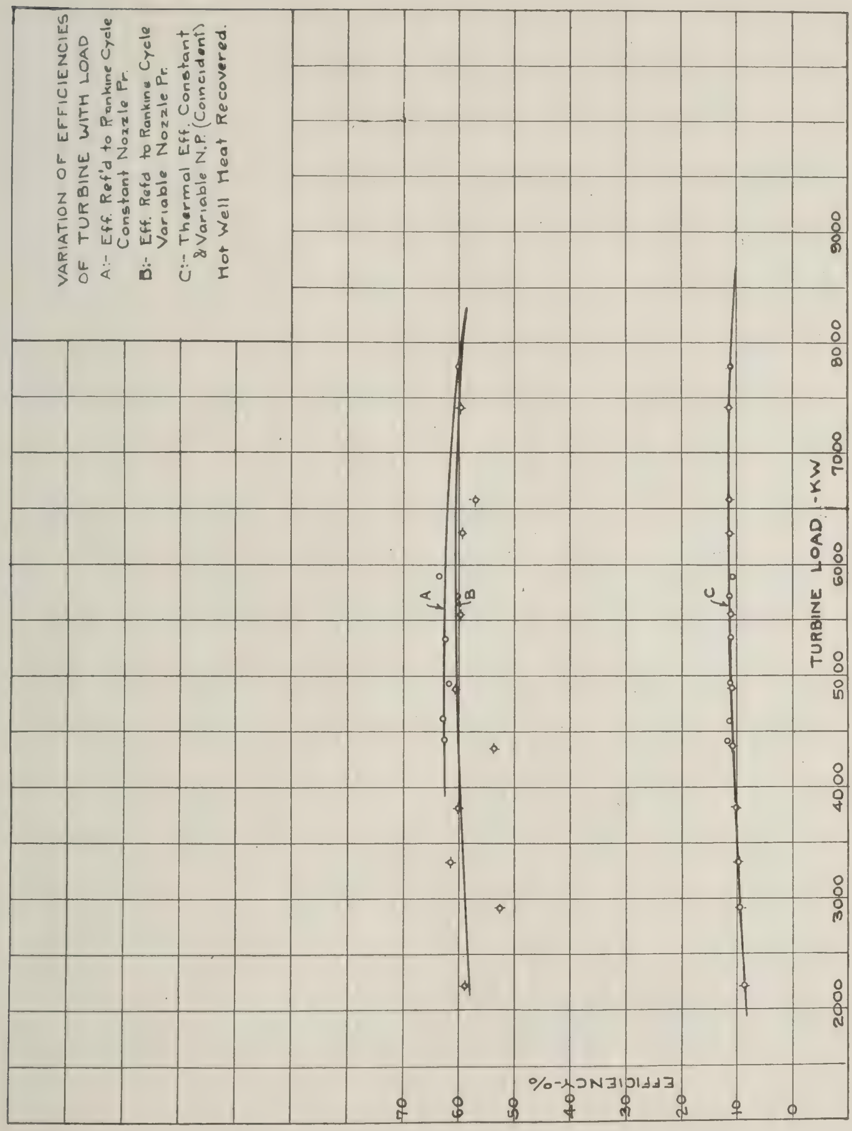


FIG. 23 SERIES E AND F

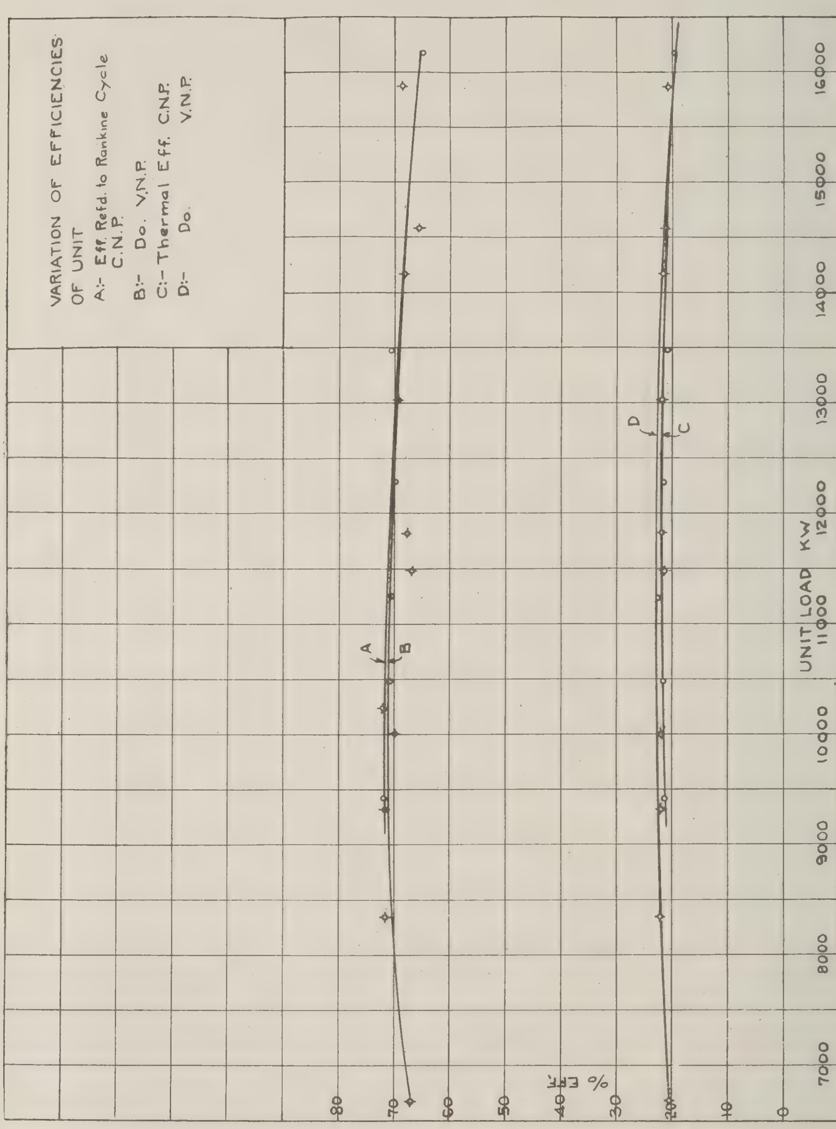


FIG. 24 SERIES E AND F

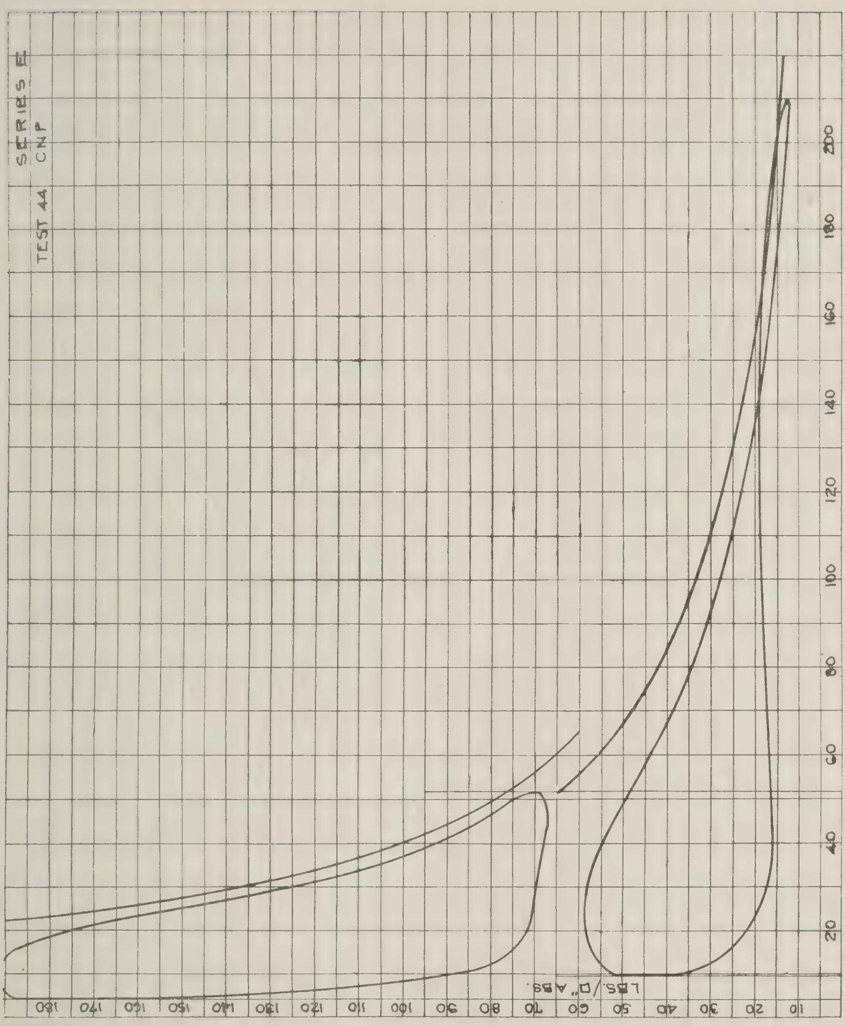


Fig. 25 Series E, Test 44

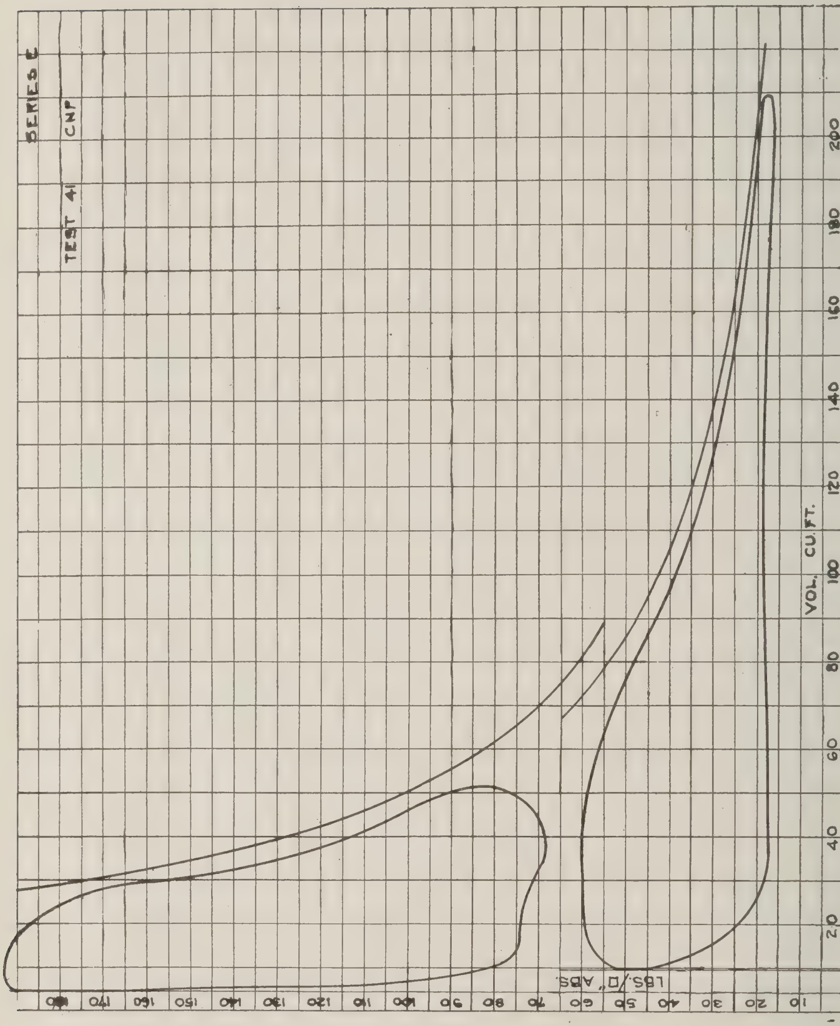


Fig. 26 Series E, Test 41

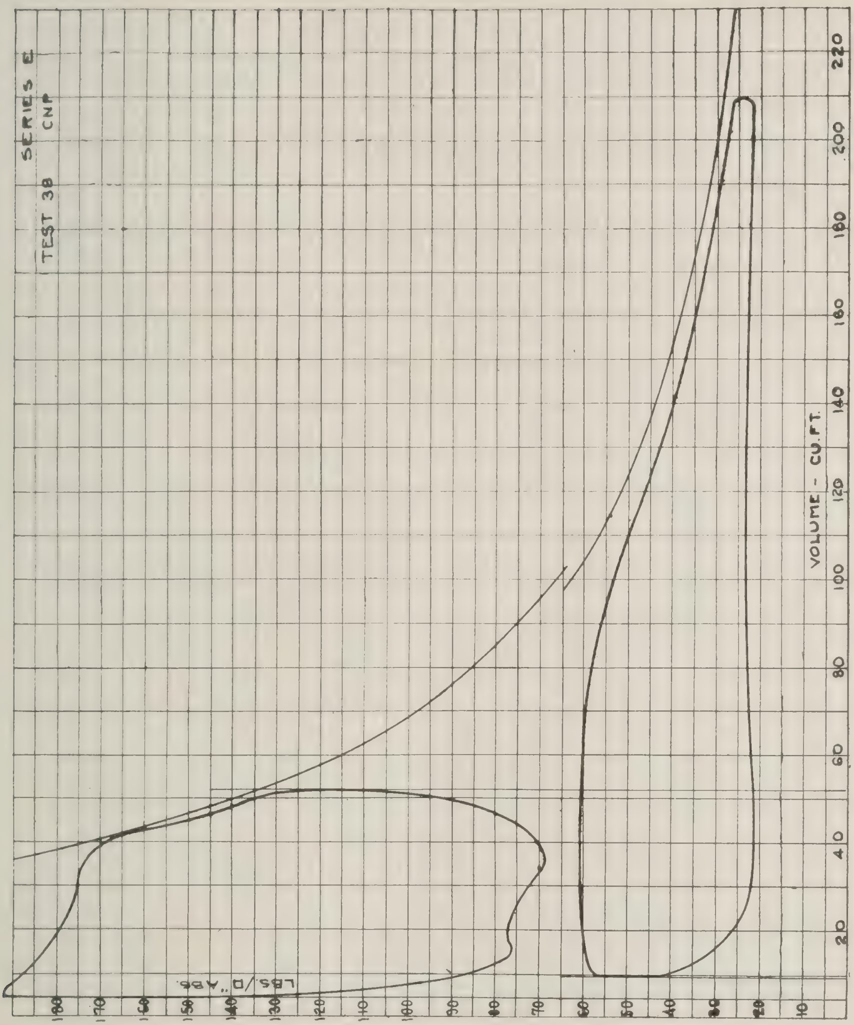


Fig. 27 Series E, Test 38

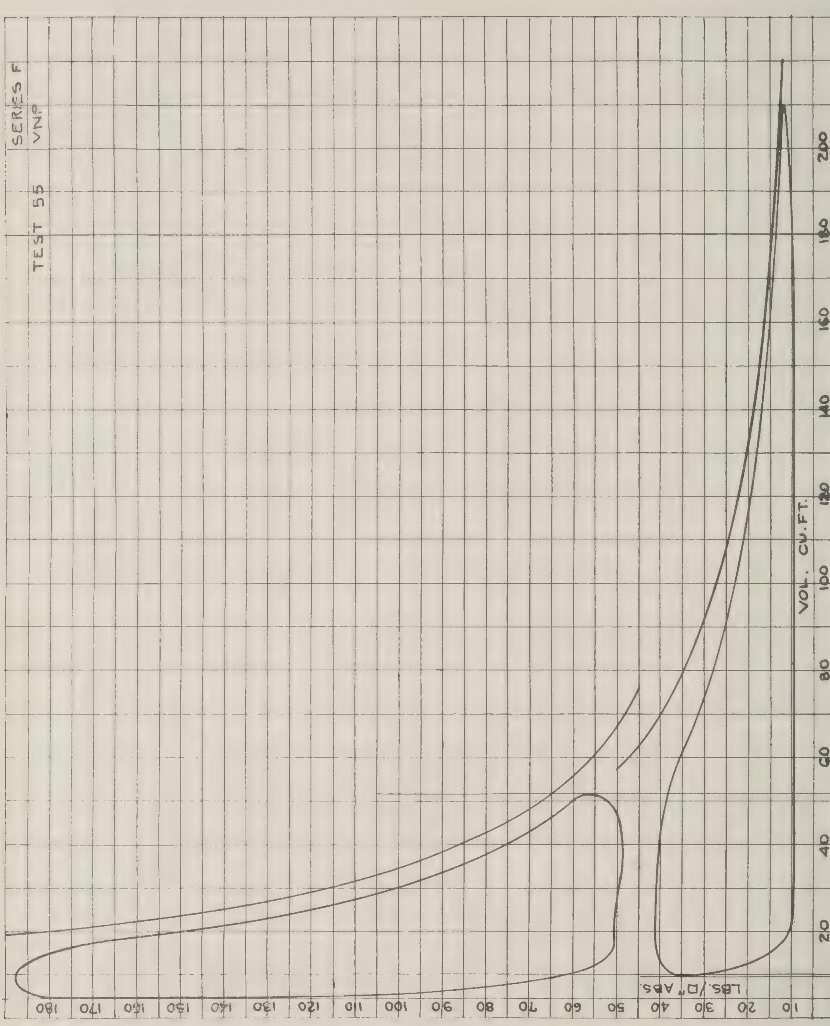


Fig. 28 Series F, Test 55

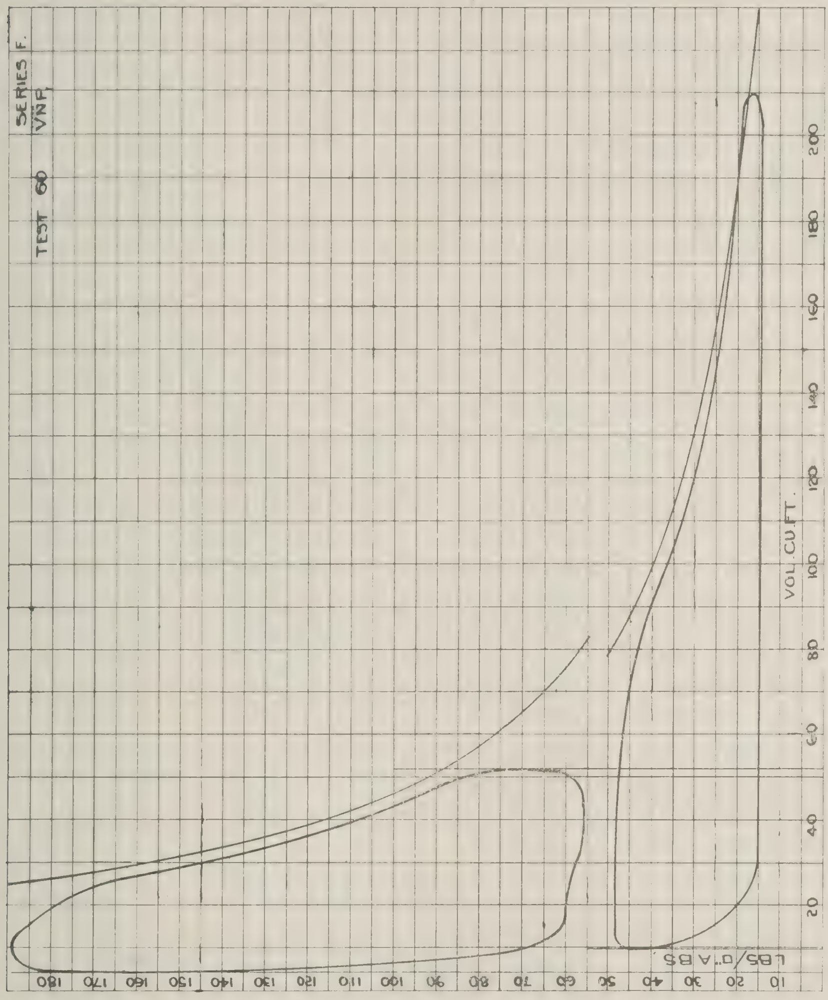


FIG. 29 SERIES F, TEST 60

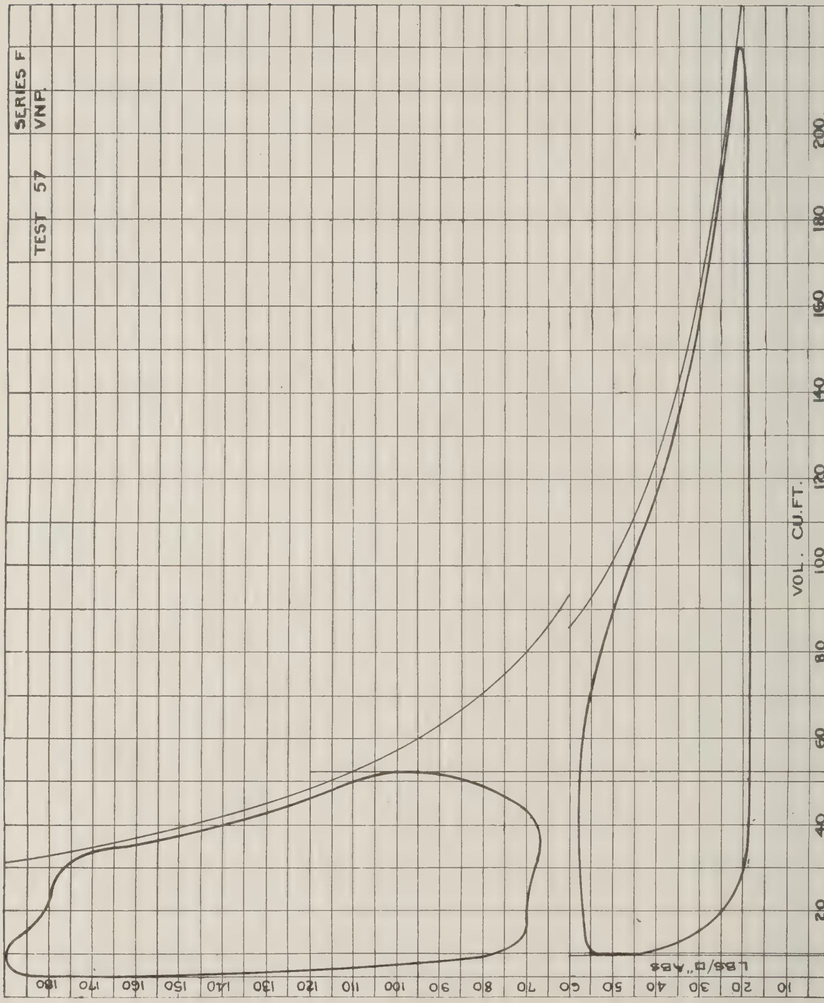


FIG. 30 SERIES F, TEST 57

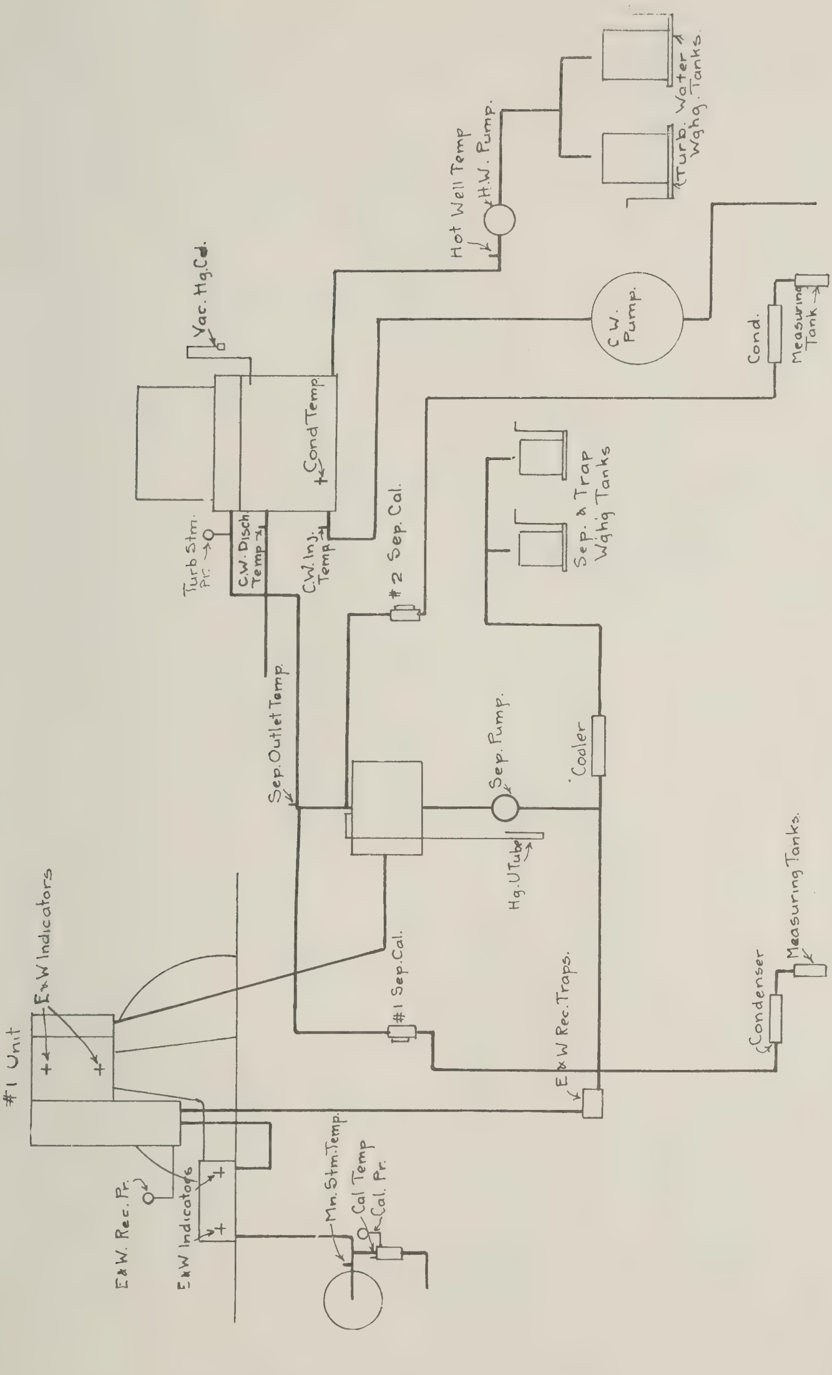


FIG. 31 SERIES C, DIAGRAMMATIC TEST LAYOUT

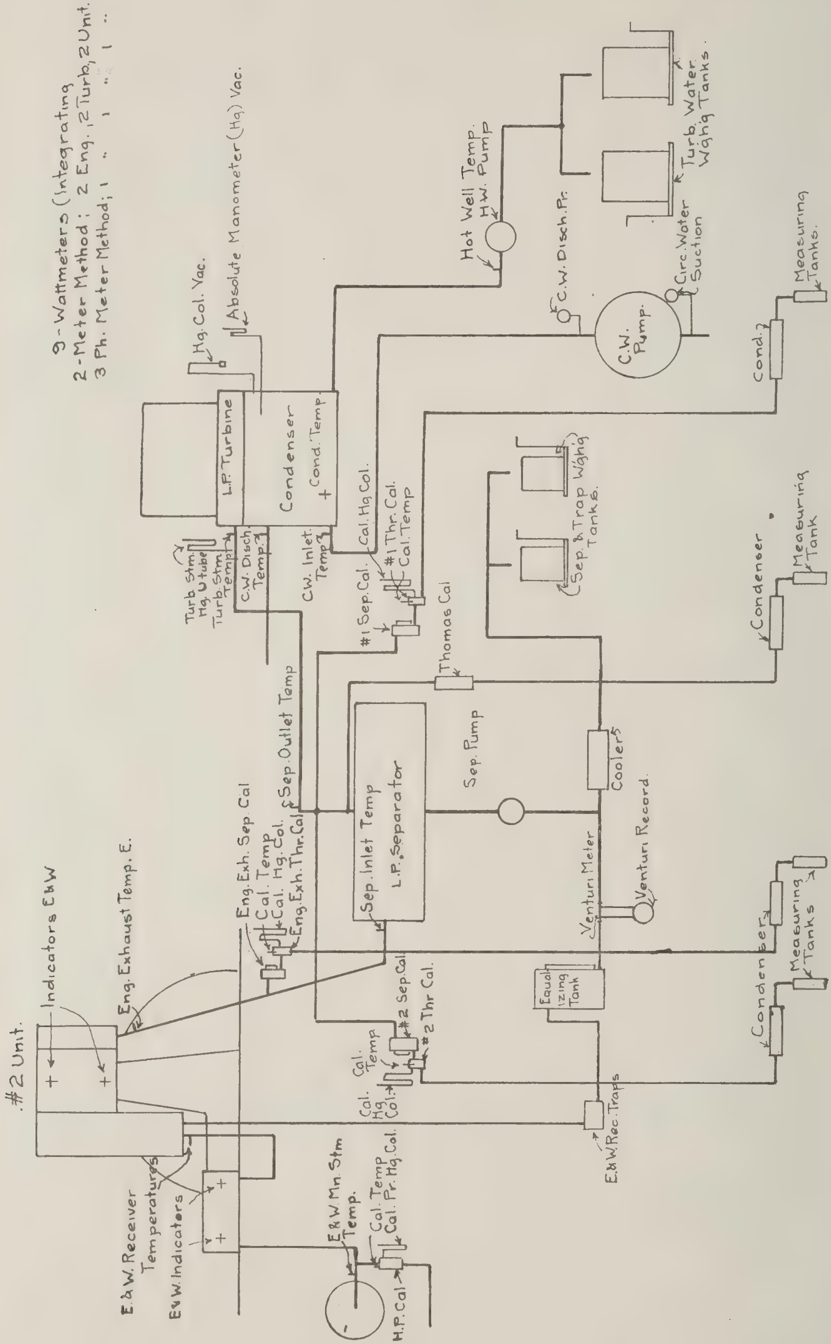


FIG. 32 SERIES D, E AND F, DIAGRAMMATIC TEST LAYOUT

ENGINE

2-H.P. CYL 42" x 60", 9" rod
2-L.P. " 86" x 60", 10" rod
R.P.M. 75.
2-14" STM. MAINS
2-16" H.P. EXH.
2-30" L.P. " .

Clearances
HP Head 9.5% Crank 10.0%
LP " 4.77 " 4.78

Avge. Total Volume H.P. 51.7 cu ft.
L P 209.9 " "

Avge. Displacement H P 47.0 " "
L P 200.3 " "

IHP Constant. HP 15.38 L P 65.57 (Avge)

All Combined Cards Worked out on Avge Basis.
Marks & Davis Tables Used for Steam Data

TABLE II

E_r = Rankine Thermal Eff., cyclic
E_t = Engine " "
H_1 = Heat in initial steam @ press. p_1, quality x (Total per Hr.)
H_2' = " " stm. @ p_2, x_2, after adiab. exp. from p_1, x_1 (Tot. p. Hr.)

$$\frac{H_1 - H_2}{H_1} = E_r \qquad E_t = \frac{KW \times 3412}{H_1} \qquad \frac{E_t}{E_r} = E_e = \text{Eng. Eff. Refd. to Rankine Cycle}$$

No Heat Recovered.

TABLE III

p_a, V_a = press. & vol. @ HP cutoff, lbs/□" & cu.ft.
p_c, V_c = " " " @ HP compression
w_a = spec. density @ p_a
w_c = " " " p_c

$V_a w_a - V_c w_c = W$, lbs. indicated stm/stroke

$$\frac{W \times 75 \times 4 \times 60}{\text{I.H.P.}} = \text{Indicated Water Rate I.W.R.}$$

$IWR(1+y) = AWR$, actual Water Rate

$y = .129(r - 4.06)$ $\qquad$ K.W. $\times 1.465$ = I.H.P.

$r = \frac{51.7}{V_a}$ for HP cyl. = Ratio of Expansion

TABLE IV

Q_e = Total Dry Stm. p. Hr. to Eng.
Q_f = Trap Water/Hr
X_e = L. P. Exhaust quality

$(Q_e - Q_f) X_e$ = Dry Stm. to Turb., Q_t

TABLE VI & VII

X_t = Quality of Stm. to Turb. after passing Separator
$1 - X_t = w$, wetness do.
K_e = Eng. K.W. Output
K_t = Turb. " "

FIG. 33 FORMULAE AND CONSTANTS

TABLE VI & VII (Cont.)

Q_e = Dry Steam to Unit / Hr.

$\frac{Q_e}{K_e+K_t}$ = Actual Water Rate, $\overline{W}$ for Unit

Q_f = Trap Water/Hr.
Q_s = Separator Water/Hr
x_1 = HP Quality

$\frac{Q_e - Q_s - Q}{x_1} = Q_t'$

$Q_t' x_t = Q_t$.

$\frac{Q_t}{K_t}$ = Actual Turb. Water Rate, $\overline{W}_t$

$\frac{Q_e}{K_e}$ = " Eng " ", $\overline{W}_e$

$\frac{Q_e}{K_e+K_t(1+w)}$ = Unit W.R., corrected for Moisture In Turb. Stm., $\overline{W}'$

$\overline{W}' - (28.5 - p_3') = \overline{W}''$ Total Corrected Unit W.R.
p_3' = Actual Vacuum Obtained, " Hg

TABLES IX & X

All Throttling Calorimeters
$x_1 = \frac{H_2 + K(T-t_2) - q}{L}$

All Separating Cals.
$x_2 = \frac{W_f}{W_m+W_f}$
W_m = Moisture
W_f = Flow

H_2 = Tot. Heat/Lb. sat. stm. @ p_2, cal. disch. pr.
x_1 = Quality
K = Sp. Ht. Superheated stm. @ p_2, T
T = temp. Sup. stm. in Cal.
t_2 = " Sat. " @ p_2
q = Heat/Lb. of the liquid
L = Latent Ht. vaporization/Lb. @ p_1

All Combination Sep.—Throt. Cals.
$x_1 x_2 = x$, Combined Quality

Thomas Electric Cal.

$$\frac{\frac{3.412E}{W} - K(T-t)}{L} = 1 - x$$

E = Watt. Hrs. Input
W = Lbs. Stm Flow
K = as above
T = " "
t = sat. stm. temp @ p
L = Latent Ht./lb. @ p
x = as above

TABLE XII, see VI, VII

TABLES XIII, XIV

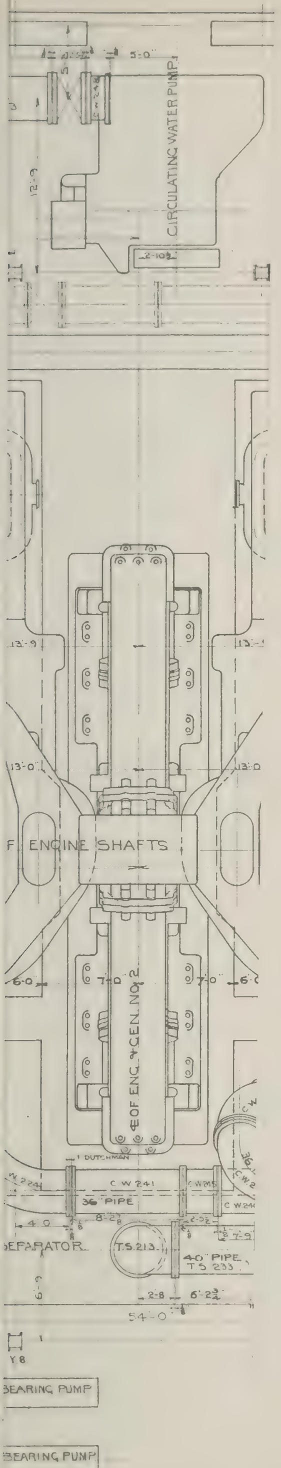
$E_t = \frac{K.W.\,3412}{H_1}$ for Eng. Turb. or Unit.

$E_t' = \frac{K.W.\,3412}{H_1 - q}$ for Unit & Turbine Only

E_t = Therm. Eff. No Heat Recovered
E_t' = " " Hot Well Heat "

FIG. 33a FORMULAE AND CONSTANTS

T



STATION, INTERBOROUG

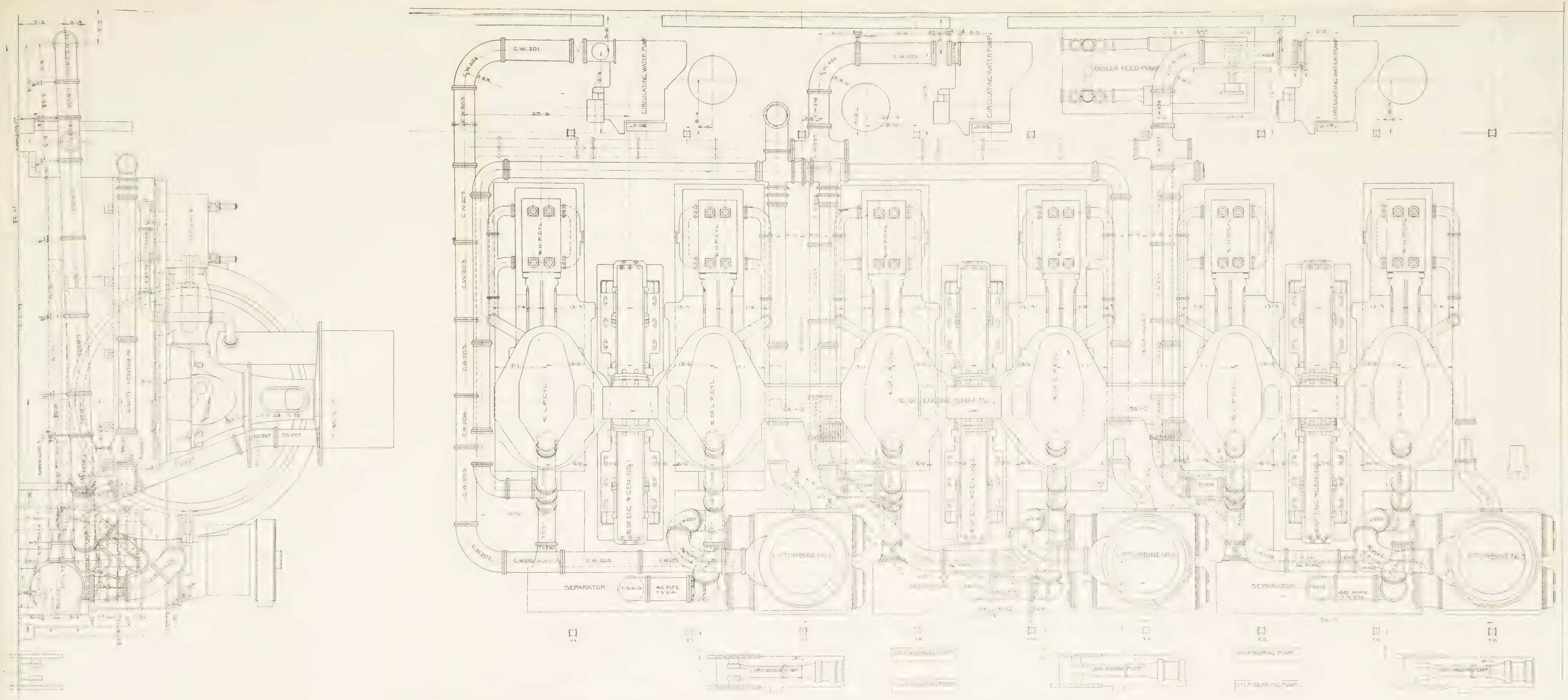
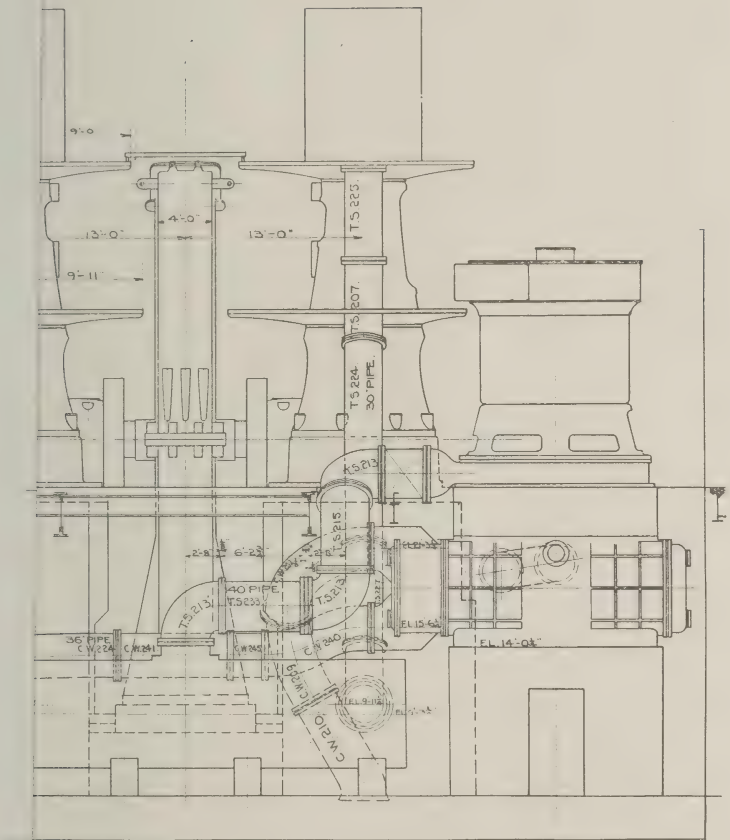


PLATE 1 END ELEVATION AND PLAN OF ENGINE AND LOW-PRESSURE TURBINE UNITS; 59TH STREET STATION, INTERBOROUGH RAPID TRANSIT COMPANY

TE TEST OF A 15,000-KW. STEAM-ENGINE-TURBINE UNIT



TRANSIT COMPANY

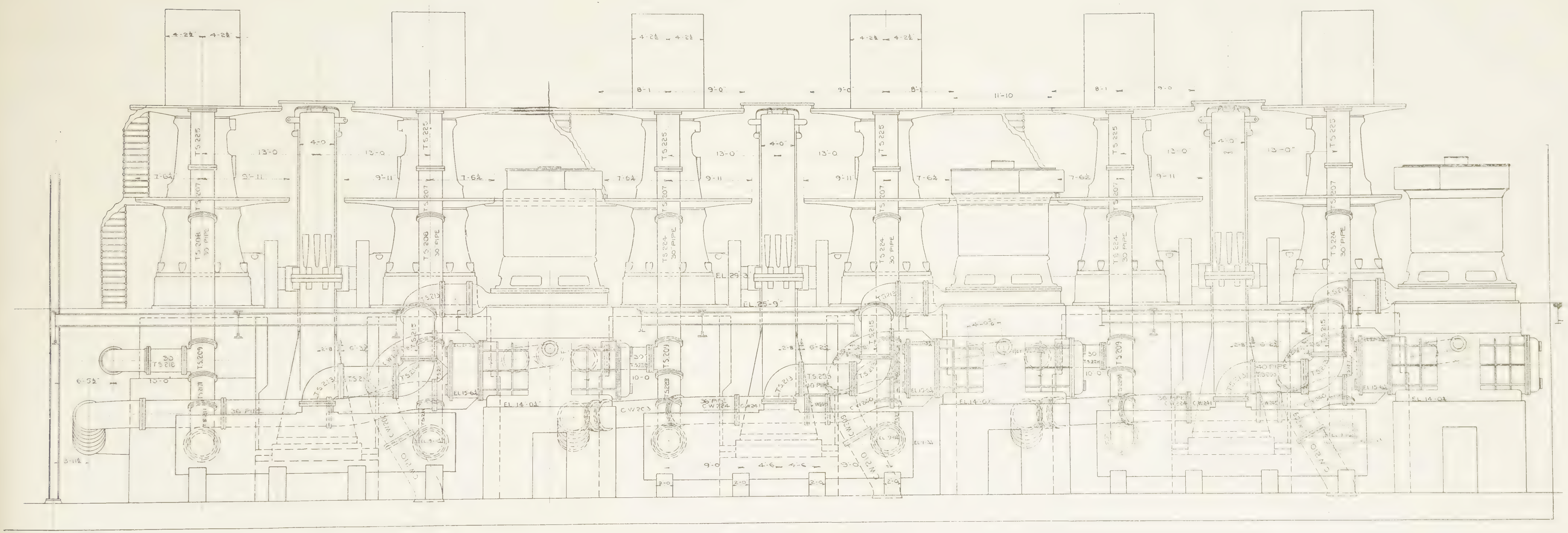


Plate 2 Side Elevation of Engine and Low-Pressure Turbine Units; 59th Street Station, Interborough Rapid Transit Company

THE ELASTIC LIMIT OF MANGANESE AND OTHER BRONZES

By J. A. Capp, Schenectady, N. Y.

Member of the Society

To keep up with the demands upon the laboratory for more work in a given time, testing machines have been speeded up and the slow extensometer has largely been displaced by the dividers, used either unchanged or with some means of magnification. To represent castings and forgings the short test piece with one-half inch diameter and two-inch gage length is almost universal. As a consequence, while reports of tests usually include a statement of "elastic limit," the property of the material actually determined is in reality that more or less vague value called the yield point. It is the object of this paper to show that while the yield point for steel is so well marked in properly conducted tests, and bears a sufficiently definite relation to the true elastic limit to warrant the dependence placed upon it by the engineer, there is no equally well defined point found in testing bronzes, and the value commonly obtained from rapid commercial tests as the elastic limit or yield point on bronze may be quite misleading.

2 Manganese bronze was selected as the metal to be subjected to the series of tests here recorded because, of the modern alloys, it is one of the strongest and is readily obtainable in the market. It is not proposed, however, to discuss at length the properties of manganese bronze as such. This metal is used as a type and the results, so far as behavior under a tensile test is concerned, may be taken as typical of brasses and bronzes in general, at least so far as they have come under the observation of the author in some seventeen years of testing materials.

3 Specifications issued by the Navy Department for managanese

bronze, March 30, 1909, required the following approximate composition:

Copper	52 per cent
Iron	1 per cent
Zinc	46 per cent
Tin	1 per cent
Manganese	Trace
Aluminum	0.5 per cent

The specification further required:

Tensile Strength	65,000 lb. per sq. in.
Elastic Limit	30,000 lb. per sq. in.
Elongation	15 per cent in 2 in.
Reduction of area	25 per cent

They state "the elastic limit is to be the yield point, measured by the drop of the bar."

4 Manganese bronze castings in the form of cylindrical bars about $1\frac{1}{2}$ in. in diameter by 24 in. long, were ordered from several foundries supplying this alloy; the orders were placed through the regular channels, bars of about this size being required in ordinary production. In this way it was hoped that commercial material would be obtained, such as might be expected in castings of more intricate shape. The results on these specimens, ordered without reference to intended use, checked very well with those upon samples submitted previously by the same parties, especially for the purpose of showing the qualities of their material. To indicate the effect of working upon the metal, there were also ordered two bars of the same dimensions hot-rolled to size, and two bars hot-rolled and cold-drawn. The effect of the cold drawing was lost to a great extent by the necessary turning off of the surface in preparing the specimen for test. Much of the cold-drawn metal is used in this way, however, when screw threads are required to provide means of fastening the part in place in the structure. From the bars so obtained, specimens were turned which provided a test section 1 in. in diameter by 8 in. between gage marks, and which had, for the purpose of gripping, ends $1\frac{1}{2}$ in. in diameter threaded to fit the nuts required by the testing machine.

5 Some of these specimens were pulled in the laboratory of the General Electric Company at Schenectady, some in the testing machine at the United States Arsenal at Watertown, and others in the laboratory of the Halcomb Steel Company at Syracuse. The tests

in the Halcomb laboratory were made to obtain autographic strain diagrams; the other tests were made with an extensometer.

6 In the tests with the extensometer, after the instrument had been placed, an initial load of 1000 or 2000 lb. per sq. in. was applied and the first reading taken; readings were than obtained at successive

TABLE 1 CAST MANGANESE BRONZE, MARK 9902 B

Extensometer Test

Original Diameter, 0.9995 in. Original length, 8 in.

Stress		Extensometer Readings		Mean Difference	Strain	
					Total	Unit
Actual	Per Sq. In.	Right	Left	Initial Reading		
1,570	2,000	0.0235	0.0075			
3,140	4,000	0.0245	0.0089	0.00120	0.0012	0.00015
4,710	6,000	0.0254	0.0107	0.00135	0.00255	0.00032
6,275	8,000	0.0263	0.0122	0.00120	0.00375	0.00047
7,845	10,000	0.0272	0.0136	0.00115	0.00490	0.00061
9,415	12,000	0.0282	0.0151	0.00125	0.00615	0.00077
10,985	14,000	0.0293	0.0165	0.00125	0.00740	0.00092
12,555	16,000	0.0306	0.0181	0.00145	0.00885	0.00111
14,120	18,000	0.0320	0.0198	0.00155	0.01040	0.00130
15,690	20,000	0.0342	0.0219	0.00215	0.01255	0.00157
17,260	22,000	0.0370	0.0246	0.00275	0.01530	0.00191
18,830	24,000	0.0414	0.0279	0.00385	0.01915	0.00239
20,400	26,000	0.0456	0.0321	0.00420	0.02335	0.00292
53,040	67,600	Tensile Strength				

Reduced diameter............ 0.695 in.
Reduction of area............51.6 per cent
Length after test.............10.20 in.

Elongation...................27.5 per cent
Elastic limit (from curve) 15,000 lb. per sq. in.
Modulus of elasticity 12,900,000 lb. per sq. in.

Commercial Test

Original diameter............. 0.503 in.
Original length................ 2 in.
Reduced diameter .... 0.387 in.
Length after test.............. 2.65 in.

Reduction of area...........40.8 per cent
Elongation..................32.5 per cent
Rapid stretch (yield point).26,000 lb. per sq. in.
Tensile strength.............69,650 lb. per sq. in.

loads applied in equal steps. In some cases, the readings were continued regularly until the increase in extension per increment of load was so great that there was no doubt that the strain diagram had departed markedly from the straight line demanded by Hook's law; in other tests, the normal succession of readings was continued only

TABLE 2 CAST MANGANESE BRONZE, MARK 9902-A

Extensometer test

Original diameter 0.995″.
Original length 8.″.

Stress		Extensometer Readings		Mean Difference	Strain	
Actual	Per Sq. In.	Right	Left		Total	Unit
1,570	2,000	0.0255	0.0125	Initial Reading		
3,140	4,000	0.0270	0.0133	0.00115	0.00115	0.00014
4,710	6,000	0.0282	0.0147	0.00130	0.00245	0.00031
6,275	8,000	0.0290	0.0163	0.00120	0.00365	0.00046
7,848	10,000	0.0298	0.0179	0.00120	0.00485	0.00061
9,415	12,000	0.0307	0.0194	0.00120	0.00605	0.00076
10,985	14,000	0.0318	0.0208	0.00125	0.00730	0.00091
12,555	16,000	0.0328	0.0223	0.00125	0.00855	0.00107
14,120	18,000	0.0342	0.0239	0.00150	0.01005	0.00126
1,570	2,000	0.0264	0.0122	0.0003	set	....
14,120	18,000	0.0345	0.0232	....	....	....
15,690	20,000	0.0360	0.0253	0.00180	0.01185	0.00148
1,570	2,000	0.0270	0.0125	0.00075	set	
15,690	20,000	0.0365	0.0250	....	....	....
17,260	2,2000	0.0384	0.0273	0.00210	0.01395	0.00174
18,830	24,000	0.0412	0.0301	0.00280	0.01675	0.00209
1,570	2,000	0.0291	0.0140	0.00265	set	....
18,830	24,000	0.0418	0.0296	....	....	....
20,400	26,000	0.0446	0.0332	0.00320	0.01995	0.00249
53,480	68,160	Tensile Strength				

Reduced diameter..0.717 in.
Reduction of area..48.5 per cent
Length after test..10.22 in.
Elongation..27.8 per cent
Elastic limit (from curve)..16,000 lb. per sq. in.
Modulus of elasticity..13,000,000 lb. per sq. in.

Commercial Test

Original diameter..0.5028 in.
Original length..2 in.
Reduced diameter..0.338 in.
Length after test..2.84 in.
Reduction of area..54.8 per cent.
Elongation..42.0 per cent
Rapid stretch (yield point)..25,000 lb. per sq. in.
Tensile strength..67,940 lb. per sq. in.

until the first positive increase in extension per increment of load was noted, when the stress was reduced to the initial load for the measurement of permanent set, after which the load was returned to the value just left, a new reading taken and the test continued with further determinations of set intervals. The values of stress and corresponding strain obtained were plotted, and the elastic limit recorded as the stress at the point of inflexion of the curve drawn through the points.

7 The specimens subjected to these tests were about 12 in. long over all. From the remainder of the 24-in. bars, the usual ½ in. by 2

TABLE 3 EXTENSOMETER AND COMMERCIAL TESTS

EXTENSOMETER TESTS: ALL SPECIMENS 1 IN. (APPROXIMATE) DIA. BY 8 IN. LONG.

Mark	9908B	Q2992A	Q2992B	Q6434A
Sample	Cast	Hot-Rolled	Hot-Rolled	Cold-Drawn
Reduction of area, per cent	14.1	52.2	52.2	53.3
Elongation, per cent	6.25	33.5	33.75	31.0
Elastic limit, lb. per sq. in	19,400	18,000	18,000	17,000
Tensile strength, lb. per sq. in	62,070	71,800	71,640	71,620
Modulus of elasticity, lb. per sq. in	13,810,000	13,000,000	13,810,000	12,800,000

COMMERCIAL TESTS: ALL SPECIMENS 0.5 IN. (APPROXIMATE) DIA. BY 2 IN. LONG.

Sample	Cast	Hot-Rolled	Hot-Rolled	Cold-Drawn
Reduction of area, per cent	28.5	44.9	45.9	39.9
Elongation, per cent	26.5	37.5	36.5	34.0
Yield Point (rapid stretch), lbs. per sq. in	29,000	30,000	30,000	43,000
Tensile strength, lbs. per sq. in	80,420	74,780	74,480	74,260

Bar 9908B was unsound, hence ½ in. by 2 in. test was turned from side of bar, instead of center. Unsoundness due to oxidation and perhaps segregation, probably accounts for the apparent cold shortness of the 1 in. by 8 in. test piece. Fracture occurred in a flaw, while many incipient fractures or cracks were noted in surface before final rupture.

in. test pieces were turned and tested in the customary commercial way, using a pair of multiplying dividers to indicate the point of increase in rate of stretch or yield point.

8 In Table 1 are given in detail a typical set of readings taken in a test at regularly increasing loads, together with the results of the commercial test upon the specimen from the same bar. In Table 2, similar data are given from a test with measurements of set. Table 3 shows the results obtained upon the other specimens tested at Schenectady. The curves for all these tests are assembled on Fig. 1. Details of the tests at the Watertown Arsenal are stated in Tables 4

and 5, and the curves from these data are given on Fig. 2. The results of the work at the Halcomb laboratory are shown in Table 6 and Fig. 3; the scale of the diagram is so small that the location of the point of inflexion is uncertain within about 2000 lb. actual load, and the values in the tables are placed rather high. The multiplying dividers used in the commercial tests here recorded magnify the movement of the gage marks about ten times, and are a much more sensitive instrument than the machinists' dividers for locating the yield point; hence, the yield points recorded are lower than are usually reported.

9 In the text books and elsewhere, the limit of elasticity is defined

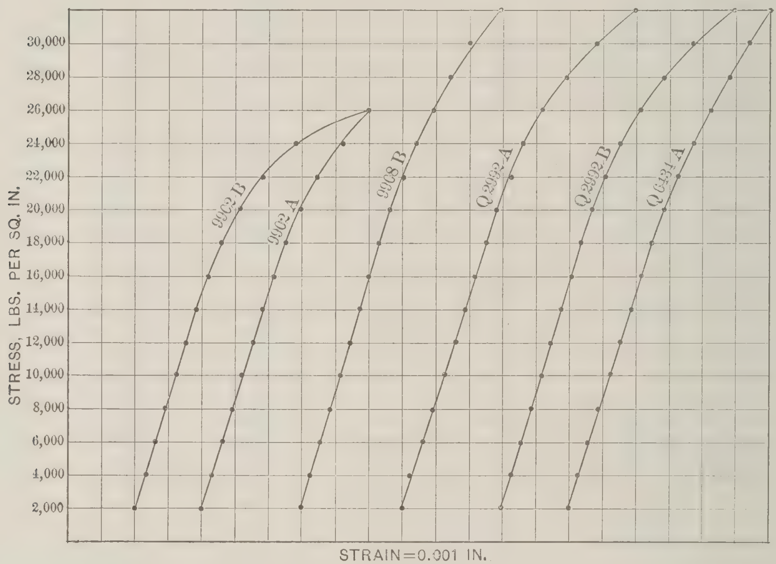


Fig. 1 Curves Plotted from Data in Tables 1, 2 and 3

as that value of stress beyond which there is not full recovery of the initial dimensions or shape of the specimen after release of the load, or as the maximum stress that can be applied without producing permanent set. In other words, it is the value of stress beyond which Hook's law no longer holds, and it is sometimes spoken of as the limit of proportionality of stress to strain.

10 Accepting this definition of elastic limit, it is seen that its value in the bronzes tested is from 16,000 lb. per sq. in. to 23,000 lb. per sq. in., whereas the yield points found for the cast metals ran from 25,000 lb. to 29,000 lb., and for the worked metals, from 30,000

TABLE 4 CAST MANGANESE BRONZE, MARK 9902 B

WATERTOWN ARSENAL TEST

Original Diameter, 1.000 in. Original length, 8 in.

Stress		Reading	Difference	Strain		Set
Actual	Per Sq. In.			Total	Unit	
785	1,000			Initial Reading		
1,571	2,000	0.0008	0.0008	0.0008	0.00010	....
2,356	3,000	0.0013	0.0005	0.0013	0.00016	....
3,142	4,000	0.0020	0.0007	0.0020	0.00025	....
3,927	5,000	0.0027	0.0007	0.0027	0.00034	0
4,712	6,000	0.0033	0.0006	0.0033	0.00041	....
5,498	7,000	0.0038	0.0005	0.0038	0.00048	....
6,283	8,000	0.0044	0.0006	0.0044	0.00055	....
7,069	9,000	0.0051	0.0007	0.0051	0.00064	....
7,854	10,000	0.0060	0.0009	0.0060	0.00075	0
8,639	11.000	0.0067	0.0007	0.0067	0.00084	....
9,425	12,000	0.0075	0.0008	0.0075	0.00094	....
10,210	13,000	0.0080	0.0005	0.0080	0.00100	....
10,996	14,000	0.0086	0.0006	0.0086	0.00108	....
11,781	15.000	0.0090	0.0004	0.0090	0.00113	0
12,566	16,000	0.0105	0.0015	0.0105	0.00131	....
13,352	17,000	0.0119	0.0014	0.0119	0.00149	0.0005
14,137	18,000	0.0133	0.0014	0.0133	0.00166	....
14,923	19,000	0.0145	0.0012	0.0145	0.00181	....
15,708	20,000	0.0167	0.0022	0.0167	0.00209	0.0025
19,635	25,000	0.0280	0.0113	0 0280	0.00350	0.0085
23,562	30,000	0.0487	0.0207	0.0487	0.00609	0.0241
31,416	40,000	0.2090	0.1603	0.2090	0.02613	0.1730
50,600	64,458	Tensile Strength				

Reduced diameter.. 0.70 in.
Length after test ....10.53 in.
Reduction of area..51.0 per cent.
Elongation..31.6 per cent.
Elastic limit..16000 lb. per sq. in.
Modulus of elasticity, 12,390,000 lb. per sq. in.

COMMERCIAL TEST (MADE AT SCHENECTADY)

Original diameter..0.503 in.
Original Length..2 in.
Reduced diameter..0.387 in.
Length after test..2.65 in.
Reduction of area..40.8 per cent.
Elongation..32.5 per cent.
Rapid stretch (yield point) ....26000 lb. per sq. in.
Tensile strength...69650 lb. per sq. in.

TABLE 5 CAST MANGANESE BRONZE, MARK 9908B

WATERTOWN ARSENAL TESTS

Original diameter, 0.7854 in. Original length, 8 in.

Stress		Reading	Difference	Strain		Set
Actual	Per Sq. In.			Total	Unit	
785	1000	0	Initial Reading			
1571	2000	0.0007	0.0007	0.0007	0.00009	....
2356	3000	0.0013	0.0006	0.0013	0.00016	....
3142	4000	0.0017	0.0004	0.0017	0.00021	....
3927	5000	0.0023	0.0006	0.0023	0.00029	0
4712	6000	0.0028	0.0005	0.0028	0.00035	....
5498	7000	0.0036	0.0008	0.0036	0.00045	....
6283	8000	0.0040	0.0004	0.0040	0.00050	....
7069	9000	0.0047	0.0007	0.0047	0.00059	....
7854	10000	0.0052	0.0005	0.0052	0.00065	0
8639	11000	0.0059	0.0007	0.0059	0.00074	....
9425	12000	0.0064	0.0005	0.0064	0.00080	....
10210	13000	0.0070	0.0006	0.0070	0.00088	....
10996	14000	0.0074	0.0004	0.0074	0.00093	....
11781	15000	0.0081	0.0007	0.0081	0.00101	0
12566	16000	0.0088	0.0007	0.0088	0.00110	....
13352	17000	0.0094	0.0006	0.0094	0.00118	....
14137	18000	0.0099	0.0005	0.0099	0.00124	....
14923	19000	0.0104	0.0005	0.0104	0.00130	....
15708	20000	0.0113	0.0009	0.0113	0.00141	0.0001
16493	21000	0.0120	0.0007	0.0120	0.00150	....
17279	22000	0.0126	0.0006	0.0126	0.00158	....
18064	23000	0.0135	0.0009	0.0135	0.00169	0.0003
18850	24000	0.0143	0.0008	0.0143	0.00179	....
19635	25000	0.0156	0.0013	0.0156	0.00195	0.0010
20420	26000	0.0165	0.0009	0.0165	0.00206	....
21206	27000	0.0171	0.0006	0.0171	0.00214	0.0015
21991	28000	0.0185	0.0014	0.0185	0.00231	....
22777	29000	0.0200	0.0015	0.0200	0.00250	....
23562	30000	0.0210	0.0010	0.0210	0.00263	0.0034
27489	35000	0.0325	0.0115	0.0325	0.00406	0.0118
31416	40000	0.0607	0.0282	0.0607	0.00759	0.0384
35343	45000	0.1225	0.0618	0.1225	0.01531	0.0920
53900	68662	Tensile Strength				

Reduced diameter........... 0.91 in.
Length after test............ 8.85 in.
Reduction of area .... 17.2 per cent.

Elongation.................. 10.6 per cent.
Elastic limit................. 23000 lb. per sq. in.
Modulus of elasticity, 13,300,000 lb. per sq. in.

COMMERCIAL TEST (MADE AT SCHENECTADY)

Original diameter......... 0.504 in.
Original length............ 2 in.
Reduced diameter......... 0.426 in.

Length after test.......... 2.53 in.
Elongation................ 26.5 per cent.
Rapid stretch (yield point) 29,000 lb. per sq. in.

Tensile strength........... 80,420 lb. per sq. in.

lb. to 44,000 lb. per sq. in. Had ordinary dividers been used, the values for the cast metals would have been placed between 30,000 lb. and 40,000 lb. per sq. in. The strain diagrams from the extensometer tests show the general shape of the elastic curve of the metal, and permit the accurate fixing of the point of inflexion of the curve; the autographic diagrams, however, show not only the actual shape of the curve, but also why there is the uncertainity in the locating of the yield point or point of rapid increase in rate of stretching.

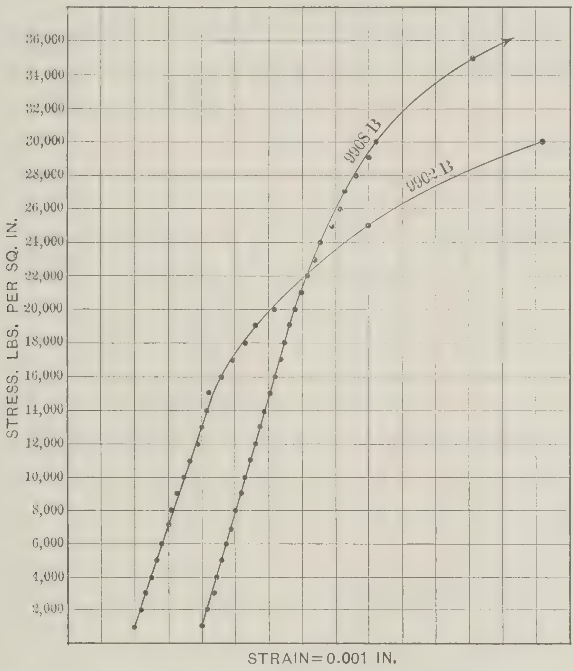


FIG. 2 CURVES PLOTTED FROM DATA IN TABLES 4 AND 5

11 For comparison, the autographic diagram of a piece of commercial "structural medium" steel is shown as No. 1 in Fig. 3. At the scale of the diagram, no inflexion of the curve is seen until it suddenly breaks sharply, actually drops and remains practically horizontal until it finally picks up again. This jog is entirely characteristic of mild steel, and is found to a more or less marked extent in all steels, save perhaps the very hard varieties. There is, however, no break of any sort in the curves obtained from bronze; they are entirely smooth. Somewhere along the knee of the curve, the tester

notes that the material is stretching faster; just where he notices it will depend upon the sensitiveness of the means employed to indicate stretch, and upon his skill and sharpness in observation. The jog in the steel curve is indicated simultaneously by the slipping of the dividers and by the dropping of the scale beam of the testing machine driven at constant speed. The scale beam does not drop when testing bronze; the operator finds the poise gradually traveling more

TABLE 6 HALCOMB STEEL COMPANY TESTS

AUTOGRAPHIC TESTS ALL SPECIMENS 1-IN. (APPROXIMATE) DIA. BY 8-IN. LONG.

Mark		9902A	9908A	Q2992A	Q6434A
Number on Curve Sheet	1	2	3	4	5
Sample	Steel	Cast	Cast	Hot-Rolled	Cold-Drawn
Reduction of area, per cent	53.6	8.3	14.1	53.2	38.6
Elongation, per cent	36.3	27.2	6.6	34.4	25.2
Elastic limit, lbs. per sq. in	38,000	21,400	22,900	22,800	25,200
Tensile strength, lbs. per sq. in	60,140	69,700	63,940	71,820	68,500

COMMERCIAL TESTS (MADE AT SCHENECTADY): ALL SPECIMENS 0.5 IN. DIA. BY 2 IN. LONG.

Sample	Cast	Cast	Hot-Rolled	Cold-Drawn
Reduction of area, per cent	54.8	22.4	44.9	39.9
Elongation, per cent	42.0	14.5	37.5	34.0
Yield point, lbs. per sq. in	25,000	26,000	30,000	44,000
Tensile strength, lbs. per sq. in	67,940	70,900	74,780	74,260

slowly to maintain balance, but who can say when the change in rate began?

12 It is customary to find the yield point in mild steels, and in fact, in annealed steels generally, at about 50 per cent of the maximum strength. The yield point in mild steels corresponds, for all practical purposes, with the elastic limit. As the steel becomes harder, due to increase in carbon or the addition of alloying metals, or to heat treatment, the yield point rises rather more rapidly than the elastic limit, although the difference between the two is not so great but that the former may be used in calculations, and the yield point itself is less sharply marked, though still observable if sufficient care is taken. The yield point in steel is accepted as a safe guide to the engineer, in deciding upon the maximum stresses that may safely be permitted in parts designed to carry load.

13 That no such dependence can be placed upon the so-called yield point, as it is determined upon bronzes, is evident; rather, recourse must be had to the slower but more accurate determination of the true elastic limit if safe data are desired. It is especially noteworthy that the sets found at the minimum values of yield point as usually reported are a very considerable proportion of the total stretch that has taken place in the metal at those stresses, and that sets are found at stresses which are but 40 to 50 per cent of these reported yield points. Under certain conditions of dead load, a stress of 75 per cent of the elastic limit is sometimes considered at least not unsafe; if such a load were calculated for bronze, upon the basis of the usual commercial test for yield point, instability of the part so designed would be inevitable.

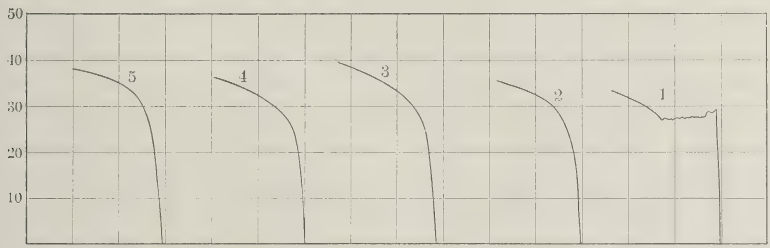


Fig. 3 Autographic Strain Diagrams to Accompany Data in Table 6

14 Hot working of the metal has not materially improved its elastic properties, but has greatly increased its toughness, and probably in an extended series of tests, would have been found to impart uniformity. It is well known that this particular alloy is relatively difficult to handle in the foundry because of its sensitiveness to temperature of pouring and to changes in composition, at least in the sense of impurities in the constituent metals, and because of its great shrinkage, requiring large feeders and sink heads. As in other copper alloys, many of the ill effects of this sensitiveness may be largely overcome by hot working. The data here presented are too meager to warrant lengthy discussion of the effects of cold working of the metal; it is shown that in the case of bars of $1\frac{1}{2}$ in. diameter, the effects of the cold drawing may have largely disappeared when $\frac{1}{4}$ in of metal is removed, except as shown in a lessened elongation. Neither hot or cold working cause any change in the elastic curve of the metal; it remains a characteristically smooth curve. In other cold-drawn copper alloys, when tested without removal of surface, the elastic curve

usually presents a much sharper bend at the knee than is found in the cast metal, or in the same metal when annealed; the same would probably be found with manganese bronze if tested as drawn, without turning. Cold-drawn metal, except wire, is seldom used without removal of the surface to provide means of fastening, and it surely is safer to test it as it is used rather than in the perhaps fictitious condition of strength due to skin hardness.

15 These results do not constitute a new discovery. In the literature of testing engineering, references may be found with direct bearing on the subject; but in these days of rapid progress and short-cut methods, much that is old, or that may be found only by search, is apt to be forgotten or overlooked. Comparatively few laboratories have autographic machines, and the use of the extensometer with a specimen only 2 in. long is not very satisfactory because of the small extension of so short a length of material under stress. Many otherwise well equipped laboratories have no extensometer. So much of experience in testing materials is based upon work done upon iron and steel that it was perhaps a natural assumption that the characteristics of these metals would also be found in bronzes and similar alloys and hence that methods of testing used successfully with one would yield equally safe results when applied to the other. Test results which are misleading are exceedingly dangerous; they induce a false sense of security which may result in the failure of structures and lead to the condemnation of a material which would be perfectly satisfactory if properly applied and not unwittingly abused.

16 The author wishes to acknowledge the courtesy of Dr. John A. Mathews, operating manager, and Marcus T. Lothrop, metallurgical engineer, of the Halcomb Steel Company, in furnishing the means of obtaining the excellent autographic strain diagrams reproduced in Fig. 3.

AN IMPROVED ABSORPTION DYNAMOMETER

By C. M. Garland, Urbana, Ill.

Member of the Society

In testing prime movers, the engineer often laments the dearth of efficient power-absorbing apparatus. Especially is this true in the testing of small high-speed machines, such as automobile engines and steam turbines. In many cases the number of machines to be tested is large, in fact in some instances each machine is given a b.h.p. test before leaving the factory; and in every case where a high degree of reliability is essential from the output, the percentage of machines undergoing test must be large. The attention of the writer was forcibly called to this need several years ago in the testing of a small steam turbine running at 2500 r. p. m., and through this experience the type of apparatus described below was designed and has been used with satisfactory results.

2 In the design of such a piece of apparatus, the following points were to be considered. These are enumerated in the order of their supposed importance.

a It should be free from binding or "seizing."

b It should be free from producing changes in the load, due to changes in the apparatus itself, such as change of temperature, wear or friction of parts, etc.

c It should be capable of absorbing and accurately indicating a wide range of loads, from zero to the full capacity of the machine.

d The regulation of the load should be positive and instantaneous.

e The apparatus should require a minimum amount of attention and be capable of continuous service.

f It should be self-contained, occupy a small amount of floor space, and be free from noise and the splashing of oil and water.

The American Society of Mechanical Engineers, 29 W. 39th St., New York. All papers are subject to revision.

g It should be capable of being quickly changed from one prime mover to another.

h It should require a small amount of cooling water.

3 In considering the above items, it will be noted that Items *a* and *b* practically eliminate mechanical-friction apparatus from the field, while Items *b*, *c* and *d* practically eliminate machines depending upon the friction or resistance of liquids for their operation. With these two classes of apparatus removed, there only remained the principle of magnetic induction for the construction of an efficient absorption dynamometer.

THEORY

4 From this principle we know that a conductor revolving in a field of variable magnetic intensity has an electric current induced

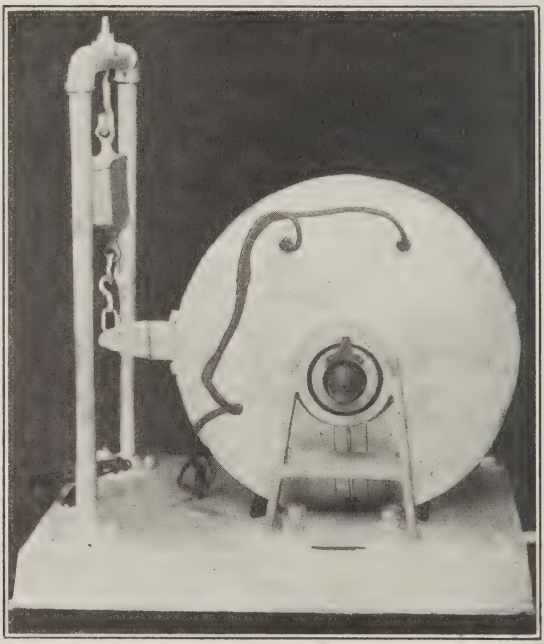


Fig. 1 Magnetic Absorption Dynamometer

in it. The reaction of this current upon the field that produces it causes a torque between the conductor and the field. There are two ways of dealing with the current induced in the conductor. In the

one, the current may be collected by a commutator or slip rings and carried off from the machine; in the other, the current, or rather currents, generated in the conductor may be allowed to remain, and, circulating in the paths of least resistance, they will ultimately short-circuit among themselves and produce heat.

5 In the first case, we have simply a dynamo mounted in a cradle. This serves as a very efficient and satisfactory type of dynamometer. There are, however, objections to its use. The currents generated must be taken care of either by water rheostats or lamp banks or utilized in the performance of work. Water rheostats and lamp banks require considerable attention and occupy space. Owing to the irregu-

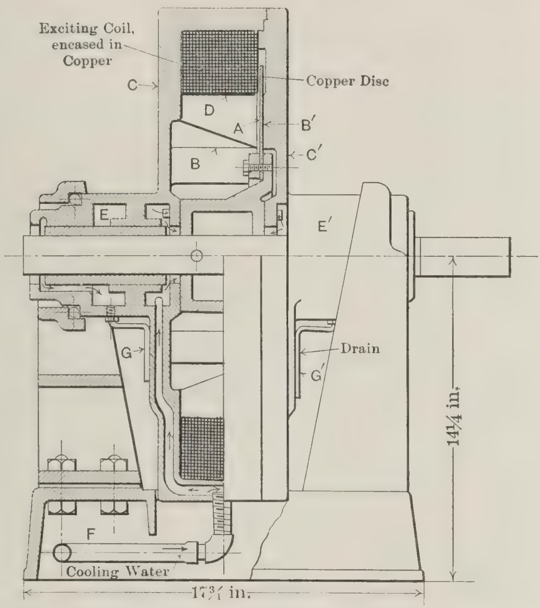


Fig. 2 End Elevation and Part Section

larities in the testing, the utilization of the current for the performance of work is in most cases impracticable. The initial cost of a testing unit of this type is necessarily large.

6 If the currents in the conductor are permitted to short-circuit themselves, the conductor is heated; the amount of heat produced is equivalent to the work absorbed by the dynamometer; and the heat thus generated may then be carried off by cooling water. This is the principle utilized in the design illustrated, a description of which follows.

DESCRIPTION

7 In brief, the dynamometer consists of a metallic disc revolving between a set of pole pieces so constructed as to produce a magnetic field of variable intensity. Fig. 1 shows the front view of a machine designed to absorb 45 h. p. at from 1200 to 1500 r.p.m. Fig. 2 is an end elevation and part section showing the construction of the dynamometer. It will be seen from this figure that it consists of a copper disc *A*, mounted on a bronze hub and revolving in front of pole pieces *B B.′* The magnetic circuit is made up of the casting *C*, the air gap and the cover plate *C′*. The castings *C* and *C′* are bolted together and carry

Fig. 3 Left Half of Field Casting Shown in Section in Fig. 2

the exciting coil *D* and the bearings *E* and *E′*. The magnetic yoke, made up of castings *C* and *C′* carrying the field coil and disc, is supported in ball bearings, and is prevented from rotating with the disc by the spring balance shown in Fig. 1. This latter measures the pull or torque between the rotating disc and the stationary yoke.

8 The magnetizing coil is encased in copper, the terminals being carried out through holes in the casting *C*, which are carefully sealed after the coil is in place.

9 The heat generated by the short circuiting of the eddy currents generated in the copper disc, is carried off by the cooling water which

enters through the base connection at F (Fig. 2) and passes up through the bearings into the field casting. It then passes out through openings which are not shown in the illustration. This water not only carries off the heat generated, but serves as a lubricant for the bearings. That which passes through accumulates in the central chamber E, and is discharged at the base of the machine through the drains G G'.

10 Fig. 3 is a detail drawing of the left half of the field casting C, shown in section in Fig. 2. It will be seen that there are six poles in the machine. The circulating water enters at I and leaves through the port at J. Similar ports are provided in the cover plate C, Fig. 2.

OPERATION

11 In operating, the engine under test is directly connected to the dynamometer shaft by means of some form of flexible coupling, the cooling water is turned on and the engine is started. After normal speed is reached, the load may be thrown on by energizing the field coil. The amount of current, and consequently the torque or pull on the spring balance is regulated by a rheostat connected in series with the coil. After running a few minutes, the quantity of cooling water is adjusted so that the temperature of the machine does not exceed 150 deg. fahr. In larger machines the coil may be wound with asbestos-covered wire and the temperature permitted to reach 212 deg., so that the cooling water is evaporated within the dynamometer. This reduces the quantity of cooling water required about 75 or 80 per cent.

12 The normal working temperature having been reached, the load on the machine remains absolutely constant, provided the line voltage is constant, for the mechanical friction, which is the bearing friction of the revolving disc, is small and practically constant, and changes in temperature due to changes in the supply of cooling water also affect the load on the dynamometer very little. The regulation by the rheostat is instantaneous and positive. When the dynamometer is driven by a smooth-running engine, the torque as indicated by the spring balance will not show a variation of $\frac{1}{8}$ lb., while the balance is sensitive to less than 1/16 lb. This indicates an accuracy that is not necessary even in the most refined testing work.

RELATION BETWEEN SPEED AND TORQUE

13 In the case of the present machine the torque is almost pro-

portional to the speed and is maximum at about 600 r. p. m. From this point the torque proPs off about 15 per cent at 1200 r. p. m., and remains almost constant from 1200 to 1500 r. p. m.

14 The torque depends upon the speed, number of poles, thickness of air gap, thickness of the copper disc, shape of the copper disc, and shape and spacing of the pole pieces. By varying the number of pole pieces, and the thickness of the copper disc, the point of maximum torque on the speed-torque curve may be shifted anywhere from 25 r. p. m. to 2500 r. p. m.

CONCLUSION

15 This type of dynamometer is well adapted either for the testing of high-speed motors with a wide variation in speed, such as the automobile engine, or for the testing of slow-speed apparatus having a small variation in the speed. It can be built in practically any size from 10 h. p. up. The principal disadvantage is the high initial cost, although this is not an item where serious and continuous testing work is going on, as in factories or in the laboratories of technical schools, for the labor saved and the increase in capacity resulting through the use of the machine will in a short time more than pay for the initial outlay.

16 The efficiency, which may be expressed as the ratio of the energy absorbed by the dynamometer, minus the energy supplied to the exciting coil, divided by the energy absorbed by the dynamometer, may be made anything up to 99.9 per cent and depends upon the weight of copper placed in the coil. Ordinarily the efficiency is made about 96 per cent, or 4 per cent of the power absorbed by the dynamometer is required in the form of electrical power for excitation.

DISCUSSION

THE HIGH-PRESSURE FIRE-SERVICE PUMPS OF MANHATTAN BOROUGH, CITY OF NEW YORK

By Prof. R. C. Carpenter, Published in the Journal for September

DISCUSSION AT ST. LOUIS

Horace S. Baker[1] presented some very complete notes on the proposed high-pressure system for Chicago, an abstract of which is given herewith. After telling of that city's need of a high-pressure system, Mr. Baker illustrated the effect of such an installation on insurance rates by citing the reductions brought about in other cities, as follows:

a Philadelphia, an initial reduction of 25 cents per $100, to be followed by a 10-cent reduction.

b Buffalo, a reduction of about 3 per cent for all buildings within 500 ft. of a fire boat pipe line.

c Manhattan Borough, New York, 10 and 15 per cent advances reduced to 5 per cent; 25 per cent advance on piers reduced to 5 per cent; storage warehouses reduced from 10 to 5 per cent; "sprinklered" risks reduced from 10 to 15 per cent.

d Brooklyn Borough, New York, 10 to 25 per cent reductions in high pressure zone.

e Coney Island, New York, 25 per cent reduction.

f Cleveland, 5 to 10 per cent reduction.

2 The costs of maintaining and operating the proposed system for Chicago should not be more than the following figures, and probably much less:

[1] Engineer, Department of Public Works, Chicago.

Operating costs of three pumping stations, including interest and depreciation	$180,000
Interest on cost of distribution system, 4 per cent of $3,000,000	120,000
Depreciation of distribution system, 2 per cent of $3,000,000	60,000
Maintenance of distribution system	50,000
	$410,000

TABLE 1 COST DATA

NAME OF SYSTEM	TYPE	PRESSURE AND CAPACITY PER MIN.	TOTAL COST EXCEPT DISTRIBUTION SYSTEM	ANNUAL OPERATING EXPENSE OF PUMPING STATIONS[1]	COST PER 1000 GAL. PER MIN. CAPACITY[1]	ANNUAL OPERATING COST PER 1000 GAL. PER MIN. CAPACITY[1]
Manhattan	Electric Centrifugal Pumps	300 lb. 30,000 gal.	$670,000	$139,250	$22,333	$4,642
Coney Island	Gas Engines Triplex Pumps	150 lb. 4,500 gal.	47,000	14,186	10,444	3,152
Philadelphia	Gas Engines Triplex Pumps	300 lb. 9,100 gal.	260,000	11,978	28,571	1,316
San Francisco Estimate 1	Steam Turbines Centrifugal Pumps and boiler Plant Oil Fuel	300 lb. 20,000 gal.	622,228	34,630	31,111	1,732
San Francisco Estimate 2	Gasolene Engines Turbine Pumps Rope Drive	300 lb. 20,000 gal.	737,848	30,595	36,892	1,529
Hartford Estimate 1	Steam Turbines Centrifugal Pumps Coal Fuel	300 lb. 12,600 gal.	257,620	45,320	20,466	3,597
Hartford Estimate 2	Gas Engines Triplex Pumps	300 lb. 12,600 gal.	377,905	8,648	29,992	686
Chicago, Estimate	Steam Turbines Centrifugal Pumps	250 lb. 10,000 gal.	263,005	37,400	26,300	3,740
Chicago, Estimate	Gas Engines Triplex Pumps	250 lb. 10,000 gal.	248,112	24,626	24,811	2,463
Chicago, Estimate	Electric Motors Centrifugal Pumps	250 lb. 10,000 gal.	122,882	57,700	12,288	5,770

[1] Exclusive of Interest and Depreciation.

TABLE 2 APPROXIMATE ESTIMATE OF COST

STEAM TURBINE PUMPING STATION, 10,000 GALLONS PER MIN. PRESSURE 250 LB. PER SQ. IN.

1	Excavation:		
	Pump pit	2300 cu. yd.	
	Boiler room	3865 " "	
	Stack	565 " "	
	Conveyor tunnel	70 " "	
		6800 cu. yd. at $1	$6,800
2	Concrete:		
	Retaining walls for pump pit	616 cu. yd.	
	Boiler room foundations	453 " "	
	Stack foundations	430 " "	
	Pump house foundation	101 " "	
		1600 cu. yd. at $7	$11,200
3	Building:		
	Pump room, 60 ft. by 54 ft. =	3240 sq. ft.	
	Boiler room, 78 ft. by 84 ft. =	6552 " "	
		9792 sq. ft.	
	Assume 10,000 sq. ft. by 30 ft. = 300,000 cu. ft. at 15 cents		45,000
4	Foundations for pumps and turbines, 150 cu. yd. at $10		1,500
5	Four 2500-gal. centrifugal pumps at $5,000		20,000
6	Four 600-h.p. steam turbines at $12,000		48,000
7	Boilers, 2400 h.p. at $15		36,000
8	Chain grates, hoppers, conveyors, etc		15,000
9	Stack		8,000
10	Suction piping from city main and tunnel		6,500
11	Discharge piping		5,000
12	Steam piping		7,500
13	Condenser		6,200
14	Boiler auxiliaries, heater, purifier, pumps, etc		9,000
15	Two 20-in. venturi meters and recorders		3,000
			$228,700
	Add 15 per cent		34,305
			$263,005

3 In the light of current practice as shown in Table 1, it seems advisable to consider and estimate on the following types of pumping stations:

a Steam turbines and centrifugal pumps.

b Electric motors and centrifugal pumps.

c Gas engines and triplex pumps.

TABLE 3 APPROXIMATE ESTIMATE OF OPERATING EXPENSE

FOR STEAM TURBINE STATION, 10,000 GAL. PER MIN. AT 250 LB. PRESSURE

1	Interest, 4 per cent of $263,005		$10,520
2	Depreciation, 4 per cent of $263,005		10,520
3	Coal:		
	200 hr., 5 tons at $2.50 8560 hr., ½ ton at 2.50		13,200
4	Oil, waste and supplies		1,500
5	Repairs		2,500
6	Labor:		
	Men, cost per annum three 8 hr. shifts:		
	1 engineer	6600	
	1 oiler	4500	
	1 fireman	3000	
	2 coal passers	5400	
	1 janitor	700	20,200
	Total		$58,440

4 For the purpose of estimate it seems proper to assume a station of a capacity of 10,000 gal. per min. against 250 lb. pressure, the working pressure to be probably 150 to 200 lb. To avoid the crippling of a station by the shutdown of any unit it seems advisable to consider units of 2500 gal.

5 In discussing the various types of installations proposed, Mr. Baker cited the advantages of each type. The direct-acting duplex pumps are rugged and ready for immediate service, but their steam consumption is large. The independent boiler plant necessary, moreover, would be costly to build and to operate.

6 The gas-engine station has the advantage of lower first cost, and no cost for power when not in operation. Though failure of the gas supply is unlikely, gasolene could be used with a change of adjustment, or by running normally on illuminating gas with low compression, which would be somewhat uneconomical. A gas-producer plant might be installed, though this is somewhat open to the same objection as the boiler plant.

7 Though electric motors are supplied from an outside source, the large number of generating stations and feeders makes the electric supply as reliable as the gas supply. The first cost and the operating expense of an electric station are low, though the standby charge is high.

8 Connecting the system to standpipes and sprinkler system

TABLE 4 APPROXIMATE ESTIMATE OF COST

APPROXIMATE COST OF GAS ENGINE STATION, 10,000 GAL. PER MIN., 250 LB. PRESSURE

1	Excavation:		
	Retaining wall	68,400 cu ft.	
	Main pit	58,089 " "	
	Engine foundations	5,096 " "	
	Pump foundations	7,056 " "	
	Tunnel	5,496 " "	
		144,137 cu. ft.	
		= 5,339 cu. yd. at $1	$5,339
2	Concrete:		
	Retaining wall	11,520 cu. ft.	
	Retaining wall footing	23,040 " "	
		34,560 cu. ft.	
		= 1,280 cu. yd. at $7	8,960
3	Building: 82 ft. by 79 ft. = 6478 sq. ft. by 30 ft. = 194,340 cu. ft. at 15 cents		29,151
4	Foundations for pumps and engines, 450 cu. yd. at $10		4,500
5	Seven 1500-gal. triplex pumps, for 250 lb. pressure at $8900		62,300
6	Seven 300-h.p. gas engines at $10,000		70,000
7	Freight and erection		7,000
8	Suction pipes from city main and tunnel		6,500
9	Water discharge pipes		5,000
10	Gas connections		8,000
11	Air compressor plant		2,500
12	Gasolene tanks and piping		3,500
13	Two 20-in. venturi meters and recorders		3,000
			$215,750
	Add 15 per cent		32,362
			$248,112

TABLE 5 ESTIMATE OF OPERATING COST

GAS ENGINE STATION

1	Interest, 4 per cent on $248,112		$9,924
2	Depreciation, 4 per cent on $248,112		9,924
3	Gas: 200 hr. at 18 cu. ft. per h.p. at $0.85 per M		6,426
4	Labor: 3 engineers at $2200 =	$6600	
	6 asst. engrs. at 1500 =	9000	
	1 janitor at	600	
			16,200
5	Oil, waste and supplies		1,000
6	Repairs		1,000
	Total		$44,474

TABLE 6 APPROXIMATE ESTIMATE OF COST

ELECTRIC PUMPING STATION, 10,000 GAL. PER MIN. PRESSURE 250 LB. PER SQ. IN.

1	Excavation:		
	Pump pit	63,936 cu. ft.	
	Retaining wall footings	8,640 " "	
	Pump foundations	2,048 " "	
	Building wall	1,692 " "	
		76,316 " " = 2,826 cu. yd. at $1	$2,826
2	Concrete:		
	Wall of pump pit	15,264 cu. ft.	
	Footings	7,892 " "	
	Bldg. foundation wall	920 " "	
	Bldg. foundation footings	329 " "	
		24,405 cu. ft. = 904 cu. yd. at $7	6,328
3	Building:		
	Pump room, 36 ft. by 56 ft. = 2016 sq. ft.		
	Switch room, 16 ft. by 56 ft. = 896 sq. ft.		
		2912 say 3000 sq. ft. by 30 ft. = 90,000 cu. ft. at 15 cents	13,500
4	Foundations for pumps and motors, 150 cu. yd. at $10		1,500
5	Four 2500-gal. centrifugal pumps at $5000		20,000
6	Four 600-h.p. 3-phase induction motors at $10,800		43,200
7	Suction piping from city main and tunnel		6,500
8	Discharge piping and valves in station		5,000
9	Switchboard and wiring in station		5,000
10	Two 20-in. Venturi meters and recorders		3,000
			$106,854
	Add 15 per cent		16,028
	Total		$122,882

in buildings had been recommended in Chicago and is the practice in Winnipeg, Man., and Providence, R. I., and also with the gravity system in Newark, N. J., Worcester and Fitchburg, Mass. The fire systems of New York City and Philadelphia are not connected in this way. The objection to these connections is that great loss of water might result from broken pipes in the buildings. This could be avoided, however, by placing a controlling valve in a brick chamber outside the curb.

TABLE 7 APPROXIMATE ESTIMATE OF OPERATING EXPENSE

Electric Pumping Station

1	Interest, 4 per cent of $122,882	$4,915
2	Depreciation, 4.3 per cent of $122,882	5,284
3	Power bill:	
	Ready-to-serve charge, $25 per kw. = $37,500	
	$0.005 per kw. per hr., 200 hr. of full load $1,500	39,000
4	Labor, 3 shifts:	
	3 engineers at $2200 $6600	
	6 asst. engineers at 1500 9000	
	1 janitor at 600 600	16,200
5	Miscellaneous: oil, supplies, etc.	1,500
6	Repairs	1,000
		$67,899

TABLE 8 ESTIMATED COST OF PROPOSED CHICAGO SYSTEM

Mains, Valves and Hydrants

District No.	Cost
1	$477,508
2	329,321
3	152,018
4	128,457
5	109,178
6	314,569
7	82,791
8	178,420
9	146,432
10	118,916
11	113,268
12	85,852
13	75,918
14	175,811
Total	$2,488,459
Engineering and contingencies	373,269
	$2,861,728
4 stations at $250,000 =	1,000,000
	$3,861,728

No allowance made for land.

River crossings are assumed to be made as follows: (*a*) North branch in present Grand Ave. water pipe tunnel; (*b*) Main River in proposed LaSalle St. water pipe tunnel, to be built by Chicago Railways Company; (*c*) South branch in present Harrison St. water pipe tunnel.

TABLE 9 HIGH-PRESSURE FIRE SERVICE IN THE UNITED STATES

City	Estimated Population	Date of Installation	Source of Pressure	Gal. per Min.	Max. Pressure, Lbs.	Lineal Ft. of Mains	Sizes of Mains, Ins.	No. of Hydrants	Total Cost of System	No. of Acres	Cost per Acre	Connection with Building	Effect on Insurance Rates
Atlantic City	40,000	Proposed	1 Station Elec. Turb. Pumps	7,000	225	38,590	8–14	82	$187,272	306	$612	...	...
Baltimore	575,000	Proposed	Pump. Sta.	...	...	75,900	10–20	...	*397,999	360	...	Standpipes on Buildings	...
Boston	620,000	1898	Fire Boat	6,000	200	4,700	12	14	30,080	65	463	...	...
Brooklyn	1,400,000	1906	2 Pump. Sta. Elec. Turb. Pumps	32,000	300	...	8–20	...	1,384,500	1420	975	...	...
Buffalo	420,000	1897	3 Fire Boats	...	300	12,756	12	...	...	...	...	...	Reduction of $0.30 per $1000
Chicago	2,229,000	...	...	...	...	...	...	...	...	...	...	...	...
Cleveland	480,000	Constructing	2 Fire Boats to Have Pump. Sta.	10,000	300	32,524	8–20	96	*170,000	338	...	May have Connection with Auto. Sprinklers.	Reduction of $0.80 per $1000 Prop.
Coney Island	...	1905–6	1 Station Gas Triplex Pumps	3,600	150	...	8–16	...	90,000	147	612	...	Reduction of 25 %
Detroit	380,000	1893	2 Fire Boats	10,000	210	25,831	8–10	95	...	356	135	...	Probably has Prevented Increase
Fitchburg	33,000	...	Gravity†	...	180	28,250	8–16	...	50,000	346	144	Boiler Feed, Elevators and Sprinklers	Prevented Increase
Hartford	98,000	Proposed	1 Station	10,000	300	53,430	8–24	198	796,277	731.3	1089	No open Connection	...
Lawrence	75,000	1906	Gravity †	...	134	10,200	10–12	39	...	120	...	...	No change
Milwaukee	340,000	1889	3 Fire Boats	15,000	250	45,717	6–12	183	...	630	...	...	10% Reduction

Newark	290,000	1905	Gravity	3,500	165	15,000	20–30	52	135,000	303	446	Some Connection	10% Reduction
New York (Manhattan)	2,100,000	1908	2 Stations Elec. Turb. Pumps	30,000	300	....	12–24	1200	3,950,400	1430	2763	Water Curtains Provided for	No change
Philadelphia	1,500,000	1903	1 Station Gas Triplex Pumps	9,100	300	35,300	8–16	166	700,000	512	1367	None on or in Buildings	Penalty of 25% Removed‡
Providence	200,000	1897	Gravity †	3,472	116	29,000	12–24	89	143,136	358	400	5 Automatic Sprinklers	No change
Rochester	185,000	1874	2 Stations Elec. Turb. Pump, Steam Turb. Turb. Pump	9,000	140	102,960	4–20	.....	....	......	.....	Some Connection	Graded Reduction
Toronto	215,000	Constructing	1 Station Elec. Turb. Pump	14,000	300	40,000	8–20	.....	500,000	287	1742	Considering Connection	Uncertain
Winnipeg	110,000	1908	1 Station Gas Producer Gas Eng., Tripl. Pumps	10,800	300	15,840	8–20	.....	650,000	275	2364	Connection with Auto. Sprinkler	Uncertain
Worcester	138,000	....	Gravity†	.....	165	100,320	8–30	.....	....	1380	.....	Elevators	No change

* Exclusive of pumping station and equipment.

† System consists of extension of pipes from high service into district covered by low service.

‡ Board of Fire Underwriters have voted to reduce rates to the amount of 10 cents per $1000 = a total of $40,000, if extensions costing $150,000 are made to the system.

NOTE:—This table is taken from the report on the proposed high-pressure fire system for Hartford, Conn.

Edward E. Wall[1] outlined the proposed fire system for St. Louis, which contemplates the installation of six or eight 5-stage centrifugal pumps, electrically driven, at a station on Chestnut St., from which the fire service mains will radiate north, south and west. The supply for these pumps will be taken from the distribution system, a 36-in. main being laid directly from the Bissell's Point pumping station to the Chestnut St. station, and connected to the present distribution system by a number of by-passes. Connections will also be made between two 20-in. mains on Fourth and Seventh Sts., to the supply for the pumps, so that in case of failure of the 36-in. main, the pumps may be supplied from this source.

2 It would be practicable to draw the fire pump supply directly from the Mississippi River by building an intake, but this would probably cost more than the laying of the 36-in. main, and would necessitate a charter from the Government. It would also raise the question of obstructing navigation, since it would be necessary to carry the construction well out into the channel, to insure an ample supply of water. Supply from the river direct would also preclude all connection with the distribution system, as it would be unwise to risk the contamination of the city's water supply by river water.

3 The pumping capacity of the station at Bissell's Point will be over 100,000,000 gal. of water every twenty-four hours, which is more than twice the amount ordinarily consumed; the excess being sufficient to supply more than 30 fire-streams through 3-in. hose continuously, assuming 300 lb. pressure at the fire pumps.

4 The 5-stage centrifugal pumps proposed for the Chestnut St. station will have a capacity of 150,000 gal. per hr. each, against a pressure of 300 lb. per sq. in. It is proposed to connect the station with the power plants of the Union Electric Light and Power Company and the United Railways, so that two sources for power will be available.

5 The three discharge mains from these pumps will be 24 in. in diameter, the district supplied by them to be gridironed by a system of 12-in. mains laid on the enclosed streets and occasionally connected, at crossings only, by by-passes, that the breakdown of one main may not necessitate the cutting out of any other line. The pipe used will be cast iron, extra heavy, with bell and spigot joints, double-grooved. All fire-hydrant leads will be 8 in. in diameter.

6 The system will be under the ordinary distribution pressure

[1] Asst. Water Commissioner, St. Louis.

when the fire pumps are not in use, so that for small fires the hydrants will be available for use; when the fire pressure is put on the system, the check valves on the by-passes will prevent additional pressure from coming on the distribution system.

7 While the arrangement of machinery for the pumping station, and the details of operation, have not been definitely decided upon, it is possible that gas engines may be used instead of electric motors. The questions of automatically starting and stopping the pumps, maintaining the pressure during a fire, and the general details of operation of the station, as well as the minor points of weight of pipe, design of hydrants, etc., have all to be worked out. It is estimated that the cost of this system will approximate $3,000,000.

H. C. Henley,[1] speaking on the advantages of high-pressure fire systems, said that they were chiefly valuable for the numerous powerful streams which can be quickly brought into service and concentrated to advantage. For the prevention of conflagrations and for keeping serious fires from spreading, more powerful streams are needed than can be supplied by portable fire engines without considerable delay. To obtain such streams from fire engines, it is necessary to "siamese"

Pressure Required at Hydrant to Overcome Friction Loss

Hose Diameter		2½ in.		3 in.		3½ in.	
Hose lines		Single	Siamesed	Single	Siamesed	Single	Siamesed
Smooth bore nozzle		1¼ in.	2 in.	1¼ in.	2 in.	1¼ in.	2 in.
Length of hose line, ft.....	100	121	139	92	101	84.5	88
	150	139	170	99.5	113	87.5	93.5
	200	158	201	107	125	91	99
	250	176.5	232	114.5	137	94.5	104.5
	300	195	263	122	149	98	110
	400	232	325	137	173	105	121

For the 2-in. nozzle it is assumed that two hose lines of the length given are siamesed together.

two or more lines into one nozzle, requiring considerable time; and if a change in the location of engines becomes necessary, considerable time is again lost in re-assembling the hose lines.

2 The high-pressure system permits the use of hose of large diameter—3 in. and 3½ in.—and direct connection to hydrants furnishes a supply to nozzles of large area, without the necessity of siamesing

[1] Chief Inspector, St. Louis Fire Prevention Bureau.

two or more hose lines. The 2-in. nozzle is best adapted for use with high-pressure systems, this nozzle, under 75 lb. nozzle pressure, discharging approximately 1000 gal. per min. A nozzle of this area provides very effective service, as the loss of pressure, due to friction in fire hose, decreases as the area of the hose is increased. The data given in the table are derived from experiments by John R. Freeman, and show the pressure required at the hydrant to overcome friction loss in hose streams of various lengths and maintain 75 lb. nozzle pressure, the nozzle being at the same level as the hydrant.

3 High-pressure systems should be considered as auxiliary protection and there should be no attempt at abandonment of engines or other apparatus.

4 Direct connection from a high-pressure system to interior standpipes, sprinkler equipments and open sprinkler systems, should be made through siamese connections and not through direct pipe connection.

5 The inability of portable steam fire engines to furnish a stream efficient to cope with serious fires is made apparent by tests made by the engineers of the National Board of Fire Underwriters. The steam fire engines for test were picked at random from the equipment of many of the best city fire departments in the country.

Number of engines tested	102
Nominal capacity, gal	69,800
Actual capacity, gal	55,900
Percentage of efficiency	80

In many cases the efficiency of individual "steamers" is less than 50 per cent.

Edward Flad. It appears to me that a cast-iron pipe is rather dangerous for high pressure. A cast-iron pipe tested under 300-lb. pressure will often break at 75 lb. A wrought steel pipe is much more reliable, and if properly coated, should last 25 or 30 years under ordinary conditions. If steel pipe is absolutely reliable we could afford to relay it at the end of 25 years rather than to use cast-iron pipe, which is liable to break.

2 In answer to a question by Mr. Flad as to the flexibility of the joint used in Baltimore, Professor Carpenter replied that it is flexible,

in the sense that it can be laid at an angle; it is not flexible as far as change of form is concerned.

H. S. Baker asked what kind of steel pipe would be used in Baltimore, Professor Carpenter answering that it is extra heavy steel-welded pipe, $\frac{7}{16}$ in. thick, the ends being expanded into semi-spheres, an 8-in. or 12-in. pipe being expanded just enough to get a ring in it, and the whole is bolted on the outside by external bolts, very like a steam pipe.

Prof. H. Wade Hibbard. It is a fact that a cast-iron water main has been in satisfactory use in city service for twenty years, and then a piece has blown out. It seems to me that the use of cast-iron pipe should be prohibited for this special emergency purpose of fire protection on account of its unreliability. In fact, in one of the high-pressure systems using cast-iron mains, leaks have been known to take place and the pumps to run for a considerable interval, some hours, I will say, and the pressure could not be maintained under test, until it was finally discovered that the water had been pouring out into a very large excavation and flooding it, unknown to those operating the pumping station. Steel will show approaching deterioration as cast iron will not.

2 Steel pipe ought to be good for thirty years of service. That period of service should be sufficient, and cities having such pipe should then be willing to replace it, having had more reliable protection during that period of years, than cast-iron pipe could possibly give.

H. C. Henley asked if there had been any attempt made to prevent the pipes from deteriorating through electrolysis, Professor Carpenter answering that the Baltimore system is a continuous metallic structure, from one end to the other, and he believed would be thoroughly protected from electrolysis; or at least, better than by any other system.

E. E. Wall. It is a fact that actually and not figuratively, steel pipe must be handled with gloves when it is laid, because the coating has to be very carefully preserved and can hardly be repaired if it is broken in handling before the pipe is laid. This is a very serious objection to the laying of steel pipe on account of exposure to corrosion after it is laid.

W. H. Reeves. Owing to the magnitude and prominence of these plants, the pump performances should be of interest to those desiring information on centrifugal and turbine pumping machinery. The highest achievement in the art of building machinery of this class is accuracy in design. Without accuracy in design it is not possible to secure the maximum efficiencies within reach. A closely designed pump should deliver exactly its contract number of gallons against the contract pumping head, and the capacity should not run over nor under. From a pump builder's point of view the misfortune of falling short of the contract capacity needs no discussion here, but the other misfortune of running over on capacity may not be so clearly understood. One effect of running over is an overload on the motor, engine or steam turbine driving the pump, and another result is that the average efficiency of the equipment in daily operation is below what it should be, for if it runs over in capacity its maximum efficiency does not occur at its contract capacity.

2 It will be noted that each of these pumps had a contract capacity of 3000 gal. per min., against a total head of 308.66 lb. per sq. in. Table 2 shows the performances of the five pumps at the South Street pumping station. This table does not show the averages, but it will be found that each pump averaged approximately 3761 gal. per min. against a mean total head of about 313.1 lb. per sq. in. Although the head was about 5 lb. above the contract condition, the pumps exceeded the contract capacity by about 25 per cent. This, no doubt, caused the motor overload mentioned in Par. 64. The contract conditions implied 540 h.p. actually delivered, and at the guaranteed pump efficiency 770 b. h. p. would be needed. The delivered work under test was 686 h. p., and according to the test efficiency of $72\frac{1}{2}$ per cent, 946 b. h. p. was used, that is, approximately 23 per cent excess motor load.

3 There appears to be no data on tests made at the contract conditions. As the pumps were tested at a great excess in capacity it is quite probable that the efficiency would have been lowered several points if the pumps had been throttled to the agreed capacity and head. The tests as per Table 2 show about 686 h. p. delivered and 946 b. h. p., or a pump loss of 260 h. p. For a considerable range it is probably safe to assume this 260 h. p. loss to be fairly constant. Assuming this to be correct and adding this loss to the 540 h. p. delivered represented by the agreed contract conditions, would give 800 b. h. p., thus showing a pump efficiency of but $67\frac{1}{2}$ per cent. If these pumps had been accurately designed, undoubtedly they would

have shown as high efficiency under the contract conditions as was obtained with excess capacity condition.

Prof. E. L. Ohle. There seems to be quite a difference in opinion among engineers as to the reasons for the variation in efficiency of the pumps when working singly and in multiple. It seems to me that the reason is the one suggested by Professor Carpenter. It is practically impossible that all should work at the same speed, as they are independently driven. If then the pressure in the main should exceed the pressure which any pump was capable of delivering, the runner of that pump would simply revolve without delivering any water. This seems to be borne out by the experience of one pump builder, as stated by J. J. Brown.

The Author. The discussion of the paper has been so voluminous that there is really but little needed from the author. In most of the discussion additional information of value has been contributed which I am sure will be appreciated by members of the Society.

2. The difficulties in connection with an installation of the kind described in the paper, involving a complete system of piping and hydrants capable of withstanding high pressures, as well as the necessary pumping machinery, are well brought out. I think the general conclusion will be that the piping difficulties to be overcome, especially when cast iron is employed, are very serious and require special skill and the best of material. Attention has also been called to the fact that the city of Baltimore has adopted a system in which steel pipe is employed in order to overcome the difficulties due to the breakage of cast-iron pipe.

3 The discussion has disclosed the construction of several stations in which the motive power has been obtained from gas engines, and the advantages, disadvantages and expense of such installation.

4 It is pointed out that although the centrifugal pumps are capable of operation at the high efficiencies shown by the paper, yet at the lower heads at which they are frequently operated the efficiency would be less. I do not believe there is any serious commercial disadvantage because of that fact, since it is true that the cost of operation of a fire station is principally due to other items than the cost of power. A fire station is required to be, above all things, reliable, and it is of very little importance whether or not the pumping be done under the most economical conditions for the reason that the total cost of pumping is only a small portion of the operating expense.

5 It is claimed by one of the discussors that the test should have

been made by the city at the exact capacity called for and the efficiency should have been based on the result of such a test. This doubtless would have produced a lower efficiency than that obtained. In the light of the information now at hand, there would have been no injustice in such a requirement, but at the date of making the contract matters were different and such a requirement would have imposed a penalty on the builders, which would have been of no advantage to the city. The reason for that opinion is, that at the time of taking the contract the information regarding multi-stage pumps operating at high heads was quite meagre. Mr. Sando, the designer of the pumps, secured all the data he could both in this country and in Europe. The result of his investigation led him to believe that it was to the advantage of the city and of the builders to put in a pump of such capacity that it would surely meet the requirements in that respect. It was believed that this would result in a considerable increased capacity over contract requirements. The motors were designed with an equally liberal capacity so that the machine was intended, even in the beginning, to be capable of a continuous large overload. The statement that the motors showed any evidence of being overloaded is in error, possibly because a certain remark which I made was misunderstood. It strikes me that the city is the principal gainer by such a system of design and that as a consequence it owns considerable more pumping capacity than was called for in the specifications, and so far as I know, without extra cost.

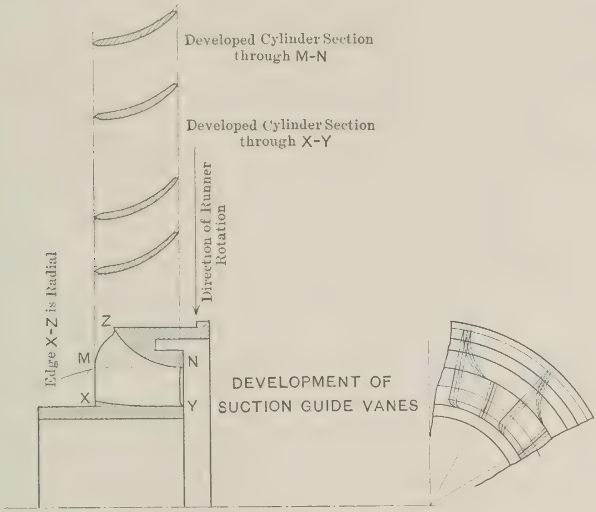
6 I believe that with the present data it would have been possible to design both pumps and motor to carry 25 per cent less load with the same efficiency as was obtained by the larger pumps and motors. In that case, a test at the specified capacity would have been a fair one.

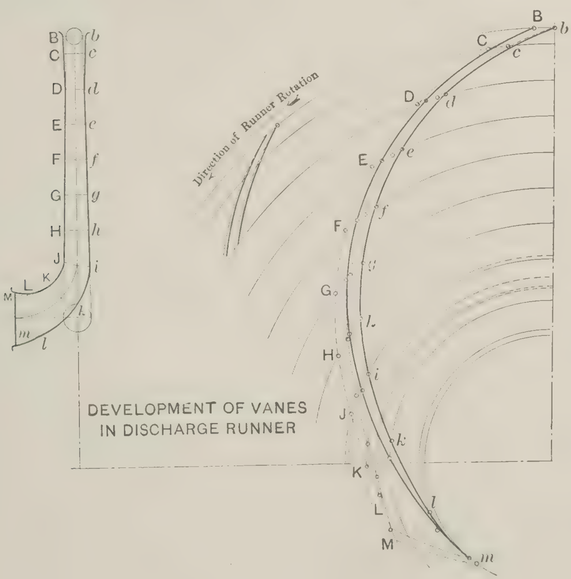
7 The interesting question brought out by these tests regarding the higher efficiency obtained with a single pump as compared with all the pumps discharging into the main, has not been satisfactorily answered. Such results, however, seem to have been noted by every engineer who has made similar tests.

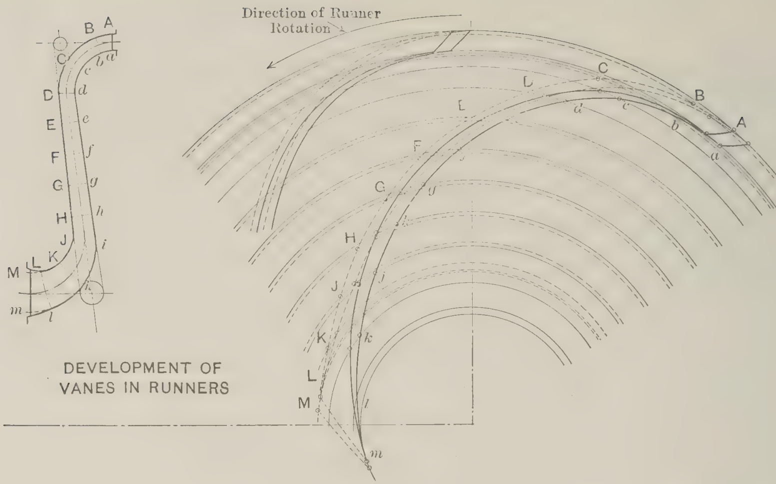
8 In the paper I made one suggestion which I believe to be of importance. I have since thought that the variation in construction or in detailed shape of the discharge volume might possibly account for some of these differences. It is hardly possible that all the pumps can be made exactly alike and small inherent differences, which would be obliterated in the operation of all the pumps together, might account for the higher efficiency of the pumps operating singly. As

suggested by Mr. White, the measurements were of a character which did not consider the pipe resistances, and the figures given apply to the delivery from the pump before the water was subjected to pipe resistances in any case.

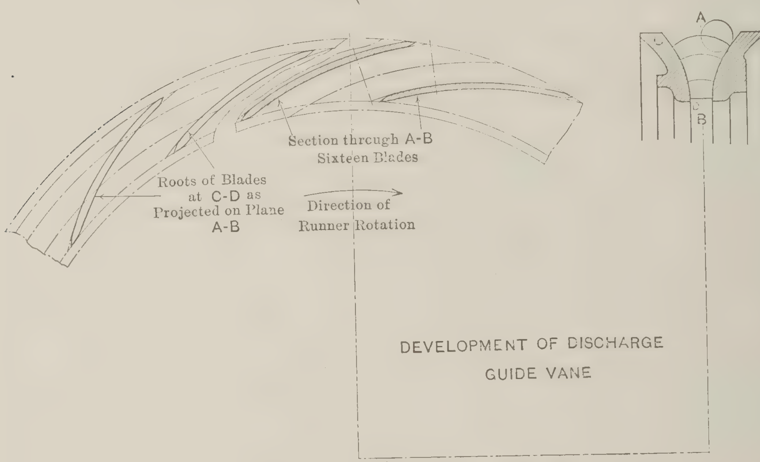
[The following curves show the development of the runners, guide wheels and guide vanes of the pumps installed in the New York high pressure pumping stations.—EDITOR.]



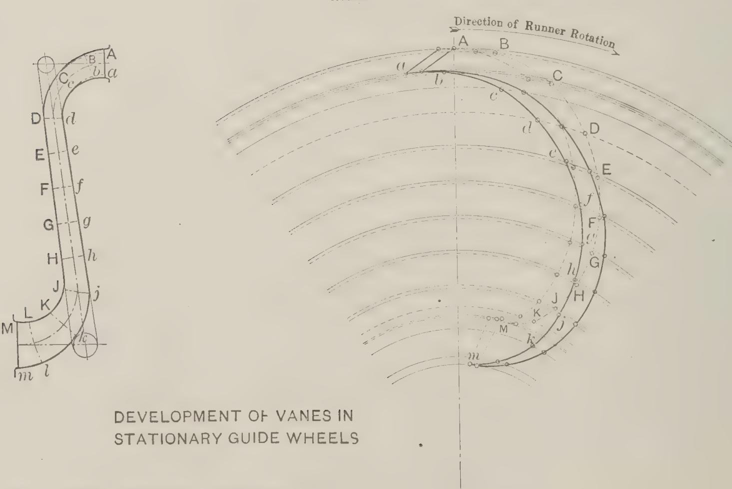




DEVELOPMENT OF VANES IN RUNNERS



DEVELOPMENT OF DISCHARGE GUIDE VANE



DEVELOPMENT OF VANES IN STATIONARY GUIDE WHEELS

LINESHAFT EFFICIENCY, MECHANICAL AND ECONOMIC

By Henry Hess, Published in the Journal for December

ABSTRACT OF PAPER

This is the description of a complete test of the relative efficiency of a lineshaft of $2\frac{7}{16}$ in. diameter, making 214 r.p.m., with bearing load due to the weight of the parts plus the tension of the belts subject to known stress by counterweight ing, when running in ring-oiling babbitted bearings and when mounted in ball bearings.

Sixteen tests, each of forty minutes' duration, with belt tensions of 20 lb. to 90 lb. per inch width of single belt, were carried out. The intruments by which the electric energy consumed was measured, as well as all other instruments and the motor, were calibrated before and after the tests. The savings in power consequent on this change ranged from 14 to 65 per cent, with 36 and 35 per cent under average conditions of good practice, due to belt tensions of 44 lb. and 57 lb. per inch width of single belt respectively.

The paper gives data for determining the power savings that may be expected in various plants as a percentage of the plain bearing shaft friction and as percentage of the total power consumption; also exact figures taking into account extra investment, depreciation, maintenance, interest on extra investment and power savings, which show that for the plant tested and described the savings represent 37 per cent per annum.

DISCUSSION

T. F. Salter. It has been long conceded that appreciable power economies were to be secured through the use of ball or roller bearings in place of plain bearings.

2 However, data which would enable the engineer to determine what savings could be expected in specific installations have been meagre as to quantity and sometimes of a questionable quality. The results obtained by the author are valuable, and are such as to encourage the use of bearings which substitute rolling for sliding friction. The following cases show the economy obtained by the use of roller bearings.

3 A Pennsylvania shoe manufacturer, with an electrically driven shop, found himself compelled to add considerable new equipment in

departments where the motors used were already overloaded. He concluded that new and large motors were necessary, but before taking action, he consulted engineers who after investigation recommended that roller-bearing hanger boxes be purchased and the old motor equipment retained. One department required 68 h.p., with babbitted boxes. The application of steel roller-bearing hanger boxes reduced the power consumption to 50 h. p., a saving of 18 h. p., or nearly 24.5 per cent, and enabled the old motors to drive the new equipment, with a small reserve for additional equipment.

4 A Baltimore belting company had a 4 7/16-in. bearing which gave a great deal of trouble through overheating. Oil bath and water jackets were tried with more or less success. A roller bearing was tried, proved successful, and forty additional bearings of various sizes were installed.

5 A wire company of Worcester, Mass., equipped their entire plant with roller bearings and have reported a 65-per cent reduction of friction load.

6 A friction disc transmission was designed by a New Jersey corporation, the requirements being that the driven shaft revolve at a constant speed. The driving shaft was subject to slight variations in speed which were to be compensated for by automatically moving the friction wheel across the face of the friction disc. The driven shaft was thus required to move laterally about $1\frac{1}{2}$ in., and to rotate at 500 r.p.m. Plain bearings with sight-feed lubrication could not be used because of their resistance to lateral motion. A special ball bearing was designed to permit a free radial and lateral movement of the shaft, resulting in an extremely sensitive and satisfactory device.

7 Roller thrust bearings are widely used wherever a thrust load or pressure parallel to the axis of a shaft is to be carried. Practically any combination of load and speed can be provided for. Nearly three years ago a bearing of this type was built for a Pittsburg steel company to operate under a pressure of 1,500,000 lb. at 100 r.p.m. As a matter of fact it carried 1,477,650 lb., applied by hydraulic pressure of 1200 lb. per sq. in. on a 32-in. piston. There was recently delivered to the same company a set of bearings the specifications of which required that they be capable of carrying 2,000,000 lb. or 1000 tons at 100 r.p.m.

8 These bearings have been applied with signal success on apparatus such as vertical hydro-electric generators, synchronous converters, frequency changers, etc., and for this work are rapidly displacing the high-pressure oil thrusts. The advantages of roller bearings are prac-

tical indestructibility, and economy of floor space (doing away with pressure pumps, accumulator, and a mass of piping required with pressure thrust); they require little attention.

9 On an installation such as a hydro-electric generating unit, it is difficult to carry on tests which would indicate by electrical instrument reading the efficiency of thrust bearings. This is due to a number of losses, the values of which it is almost impossible to determine; for instance, the loss in guide bearings, windage, variation in load on thrust bearing occasioned by fluctuations of gate openings, etc. Laboratory tests have enabled the manufacturer to be reasonably sure of the possible efficiencies which could be secured. Data obtained in this way are not as acceptable to engineers in general, however, as results secured through actual practice.

10 Believing that calculations could be made which would closely indicate the efficiency of this type of bearing, tests were made in which the rate of flow of the oil, the temperature of the oil, and the revolutions per minute of the bearing, were carefully recorded. The load was estimated and might have varied, thus affecting results. Two machines were tested, each test lasting about a week. Readings were taken at intervals of ten minutes.

11 The bearings tested carried an estimated load of 140,000 lb., at a speed of 250 r.p.m.; the temperature rise was 50 deg. fahr.; the flow of oil was $11\frac{1}{4}$ quarts (18.8 lb.) per min. From the data obtained the coefficient of friction was calculated to be 0.0016 or 0.16 of 1 per cent.

12 In the tests referred to, the heat loss, due to radiation from the oil casing of the bearing, was calculated to be 2 per cent of the total heat generated. Another test was made later with the oil casing jacketed with asbestos and the results showed a difference of 2.74 per cent.

13 These figures may be somewhat low; laboratory tests indicate that they are. I believe, however, that with a bearing of this type designed to meet the conditions of load and speed under which it is to operate, a coefficient of friction of less than 0.0025 can be obtained readily.

C. A. Graves. In tests made on something over two hundred different line shafts in various industries, I have found that a unit termed "watts per bearing" is best suited to making comparisons. This unit was obtained as follows:

2 Tests were made, stopping all the machines connected to the

shafting and measuring the power required to run the motor and shafting. The main motor belt was then taken off and the power required to run the motor free was found. The hanger bearings were counted and also the loose pulleys over which belts were passing. The difference in power, measured in watts between the shafting running free and the motor running free, was divided by the number of hanger and loose pulley bearings.

3 It developed that, on the average, loose pulleys and the hanger bearing of about the same size took approximately the same amount of power, so that the sum of the loose pulleys and hanger bearings was called the "bearings." These tests were tabulated, first, by class of industry or business, and then according to the size of the shaft. For instance, in fifty tests in machine shops, with speeds ranging from 150 to 300 r.p.m., the average power absorbed by the shaft is 49 watts per bearing. Other tests gave results shown in the table.

No. of Tests Made	Size Shaft Ins.	R.P.M.	Power Consumed Average Watts per Bearing.
43	1	400	27.1
21	1½	320–400	66.8
38	2	190–400	99.1
4	2½	200–250	108

One-inch shaft means ⅞ in. or 1$\frac{1}{16}$ in.

4 We were fortunate in having eight different shafts equipped with roller bearings and loose pulleys. It was found that with the shafts running from 108 to 300 r.p.m., 22 watts per bearing were required, with roller bearings on a 2-in. shaft. Taking the author's figures of tests, 3 *A* would give 5.25 watts per bearing, while 4 *A* would give 62.0 watts per bearing.

5 The author might have mentioned an additional saving obtained by using ball bearings, as smaller motors may be used to drive the shaft, thus reducing the fixed charges.

C. J. H. Woodbury. Without questioning the general conclusions of the author, I wish to inquire if the three per cent coefficient of friction referred to in Par. 31 was derived from his experiments or from other sources. The friction of a lubricated bearing varies according to the temperature of the bearing and the pressure upon it. Different oils also give different results. With light pressures, the vis-

cosity of the oil plays a large part, so much so that if the film of oil is thick, the internal resistance from the fluid friction among the particles of this oil constitutes a large element.

2 Under heavy pressures the film of oil becomes thinner, the resistance due to its internal viscosity becomes diminished and the frictional resistance of the whole bearing approaches a direct ratio of the pressure upon it. In other words, the coefficient of friction becomes very nearly constant and slightly diminishes with increased pressure as long as the lubrication is sufficient to prevent the material of the two surfaces from coming into contact with each other, after which the frictional coefficient increases, although it may not reach the conditions of a hot bearing.

Walter Ferris. The coefficient of friction of railway journals is extremely low. Without being sure of the accuracy of the statement, I believe it is nearly always below one-half of one per cent, and approaches one-quarter of one per cent. Under these circumstances, granting for the moment the correctness of the statement, the saving of friction due to the ball and roller bearings would have to be balanced carefully against additional complication, first cost, and delay in making repairs.

Fred J. Miller. The author has given no description or drawings of the bearings. The language of the paper will apply quite generally to ball bearings, whereas I understand that the test was made with specific ball bearings which had been in use for five years. I think we should have all the specific information about these bearings—including drawings—that the author is inclined to give, and a statement of the degree of refinement necessary in the making of the bearings in order to get these results.

Arthur C. Jackson. An advantage of ball bearings over plain bearings is that the speed of the shaft can be decidedly increased, permitting a reduction in the weight of the shaft and the driving pulleys, and reducing windage and other losses. The smaller driving pulleys will give an increased arc of contact for the belt on the driven pulley. In my experience in driving high-speed machinery, increasing the speed of the line shaft, which can be accomplished by the use of ball bearings, has a distinct advantage.

Chas. D. Parker. The value of the ball bearing or roller bearing seems to be conceded in a general way, but its application imme-

diately brings up the question of excessive cost, so that it is hardly considered in many cases. Data of the sort given in the paper should be highly valuable as giving confidence to engineers in recommending the use of ball bearings on a large scale, even though the cost may be high. The question cannot be decided by a single experiment. Several experiments, including tests on a shaft 400 or 500 ft. in length, would be even more valuable, especially if made on bearings that have had a few years' service under ordinary care.

2 It might be of interest to know whether the apparently high cost of ball and roller bearings is due to the high cost of manufacture or to large selling expense, which we may expect to be reduced with a more general demand for the goods.

3 With the general introduction of electric-motor drive, the belt drive from line shafts has become somewhat old-fashioned. However, as the motors have large factors of inefficiency, if the efficiency of the line-shaft belt drive can be greatly improved by the use of ball bearings, it would be of interest to know to what extent this can be done. It would probably be shown that the older method is still the more economical method in a great many instances.

Oliver B. Zimmerman. I would like to ask Mr. Hess if he has considered the application of ball bearings to countershafts which do not run the same proportion of time as the line shaft. What would be the relative return on the investment in that case, as compared with the line shaft itself? Furthermore, would it be advisable to lengthen the line shaft when the ball bearings are used; for instance, in group driving, would it be an advantage to use a line shaft 90 ft. or 100 ft. in length, as compared with a group of machines driven from 60 ft. of line shafting?

W. F. Parish, Jr. Mr. Hess's paper brings out an important point usually overlooked in comparative tests requiring great accuracy, namely, the influence of temperature and relative humidity on the power delivered, by causing variations in belt tension.

2 For comparative tests made under work-shop conditions it is advisable to have the belts made up half of cotton and half of leather, thereby eliminating the effect of humidity, which may cause variations of 12 per cent in the power delivered.

3 An English firm five years ago purchased a cotton belt to drive a dynamo, but this belt was not equal to the speed and power required of it, so a leather belt was substituted. It was decided to use the cot-

ton belt on one of the main mill drives, but it was found to be much too short. So a piece of leather belt was spliced in, the whole being, when finished, half leather and half cotton. A casing was built under it, as it was low down and in a dangerous position. The manager was annoyed to find that this casing had been built too close to the belt, no allowance being made for sagging.

4 The dampness greatly affected the leather belt, as the drive was in a low part of the mill, but the casing under the patched belt was never altered. The length of this belt never varies whether the weather is damp or dry and it is the best belt drive in the mill for steady work. Moisture has an opposite effect on leather and cotton, leather lengthening and cotton contracting with an increase of humidity, so that in the half-cotton and half-leather belt the weather effect is practically compensated for.

5 In Test 3 and Test 4, the average saving of power by using ball bearings instead of ring-oiling bearings is 36 per cent and 35 per cent, respectively, which is unusually good. It would be interesting to know what oil was used in the ring-oiling bearings during these tests and if the oil was new or old. With a very poor oil in the ring-oiling bearings the saving in power may be only partially caused by the change to ball bearings.

6 Oil and lubrication play a very important part in the economical distribution of power. Many power tests which I have made show that when very poor and cheap oil is used, a saving as high as 40 per cent can be obtained simply by using a better oil. Forty-two comparative power tests, made in small work shops or sections of large shops, show an average saving in power of 19 per cent, due to the use of a good and suitable oil. By using a good oil there will be but little increase in cost, as it can be used sparingly, so that the yearly cost for the better oil may be even less than for the poor oil. One test on a machine gear-driven by an electric motor showed a power saving of 55.5 per cent. by using a good oil instead of a poor oil and grease.

Geo. N. Van Derhoef. In the results of the tests summarized in Par. 41 of Mr. Hess's paper, the quantity of oil required to maintain ten 2 7/16-in bearings is given as ½ pint a day, or 150 pints per year, which is equal to 18 ¾ gal. There is probably no make of self-oiling hanger on the market today that requires anything like this quantity of oil to maintain it. Three gallons a year for ten hangers would be ample allowance for even the poorest make.

2 The item of labor charged is two hours a week, which is also

excessive even if the enormous quantity of oil specified were used. As a matter of fact, three or four hours a year should be ample time to devote to the care and attention of ten 2 7/16-in. hanger boxes.

3 The allowance of twenty years for depreciation would seem fair for babbitted bearings, as probably all of us know of bearings running in daily service for a longer period. I would like to know if Mr. Hess has any figures showing ball bearings on line-shaft service for anything like this period. As I look at the matter—and I think others will agree with me—it is not so much a matter of a lower coefficient of friction as it is of the "staying properties" under practical conditions, as distinguished from a test experiment extending over a brief interval of time.

The Author. Taking up the various points raised and the questions asked during the discussion, the author wishes to reply as follows:

2 *Percentage of Saving and Actual Saving.* A saving of power cannot be intelligently considered as a percentage of the entire driving power without full knowledge of the entire conditions. A given actual saving may be one per cent or ninety-nine per cent of a total. The saving in line-shaft journals when referred to the line-shaft loss is one ratio, and when referred to the total power consumption, is quite another ratio. So far as I am aware the literature on the subject quite generally refers to the line-shaft friction as a percentage of the total power consumption. That is misleading, since the percentage ranges from only sixteen or so in some textile mills to seventy or more in some of the rougher machine industries. In all probability the actual friction loss, bearing for bearing, does not vary in anything like so great a degree as sixteen to seventy per cent. The thing that is of real importance is not the ratio of the saving to a given whole, but the actual value of the actual saving.

3 *Estimating Power Losses and Savings.* Mr. Graves suggested that the power consumption of a bearing might be stated from experience in "watts per bearing." Such an expression would be convenient if it could be correctly applied; but the watt loss depends upon the coefficient of friction, the load and the surface speed. The coefficient of friction for a given type of bearing may be said to be fairly well known, or at least not to vary between very wide limits. That may also be said of the load; but the surface speed is made up of the shaft diameter, or rather the circumference, and the angular speed, both varying between very wide limits. So general an expression is therefore hardly possible, nor is it necessary.

4 For any given installation, the shaft diameter and speeds are known; the loads are due to the definitely determinable weight of the shaft, pulleys and belts, and to the belt pull, the last-named of which should not be allowed to exceed 60 lb. per in. width of single belt, while it certainly will rarely fall below 40 lb. The coefficient of friction for plain bearings may range from 2 to 8 per cent, with 3 per cent a very fair and general value, and $\frac{1}{8}$ per cent for ball bearings. A rise to $\frac{1}{4}$ per cent for ball bearings would indicate a poor quality of bearing.

5 An actual calculation, using the known constants of the installation in question, will always give closer results than the use of any general expression, necessarily much less accurate, such as "watts per bearing." In Par. 32 the expression for kilowatts is given as

$$Kw = 0.000,0059\ L\ d\ s\ \mu$$

or

$$\text{watts} = w = 0.000,000,0059\ L\ d\ s\ \mu$$

which may readily be converted to the convenient form

$K\,w_y$ = watts per bearing for year of 3000 hours

$K\,w_y = 0.000,001\ L\ d\ s\ \mu$

6 Mr. Graves has found the "watts per bearing" to range from 27.1 to 108 in 106 tests of plain bearings. The measured losses of the test cited in the paper are under average conditions of belt pull. For the usual belt load, Test 3 and Test 4 show for the ten plain bearings (see table in Par. 33) losses in kilowatts of 0.350 and 0.405, and for the ball bearings 0.018 and 0.020, or in watts per plain bearing 30 and 35, and for ball bearings 15 and 18.

7 Mr. Graves' four tests of a 2 7/16-in. line shaft at 200 to 250 r.p.m. may be fairly compared with the author's tests of a 2 7/16-in. line shaft at 214 r.p.m; Mr. Graves' result of 108 watts per bearing, as against the author's of 30 to 35, shows how unsafe a general wattage figure is. Changing the coefficient of friction from the 3 per cent found to be approximately correct for the test cited, to 10 per cent, would raise the 30 watts per bearing to Mr. Graves' 108 watts per bearing. In reality the tests cited by Mr. Graves are confirmatory of the author's, since the former range from 27 to 108, proving that the author's values of 30 to 35 for correct belt loads and 22 to 46 for extremely light and extremely heavy loads, represent an average of good practice.

8 *Indirect Economies.* Mr. Graves has suggested that the mounting of line shafts on ball bearings will reduce the sizes of the

motors required to drive the shafts. While that is obvious, the consequent economy is greater than is at first apparent. A motor must always be selected of sufficient size to perform its work safely. As the frictional resistance of a plain bearing line shaft is apt to vary between very wide limits—27 to 108 watts per bearing, according to Mr. Graves' tests—the motor must necessarily be selected to cover nearly the maximum safety. That means a rather large motor compared with the average useful plus friction load. Not only is there thus an unnecessary increase of first cost of the motor but, more seriously, the operating cost is unduly enhanced, as it is well known that a motor operating much below its rated capacity has low efficiency and is wasteful of current. When, on the other hand, the line shaft is mounted on ball bearings, the friction load is greatly reduced, its amount is more definitely determinable beforehand, and the initially smaller motor is used nearer at its point of maximum efficiency.

9 Mr. Jackson refers to a possible increase in shaft speed due to mounting the shaft on ball bearings, resulting in decreased weight of shaft pulleys and belts and more favorable belt contacts. All of these elements in time make for decreased bearing loads and consequently still further increases in economy. Mr. Jackson has had under his continual observation during several years a number of plain and ball-bearing line shafts of medium and high speeds, and so speaks not merely from theoretical reasoning, but from actual practice and observation.

10 *Reduced Importance of Improper Belt or Rope Tension.* The great variations in belt tensions that may be brought about by weather and temperature conditions, moist and dry atmosphere, etc., have been referred to by Mr. Parish. Both leather and cotton belts, as well as fibre ropes, are subject to considerable variations from these conditions. Possibly fully as important a factor is the average millwright or mechanic. The properly stressed belt is the exception. Most belts are tightened almost to the breaking point. The work thus lost in friction in plain bearings is directly proportional to a coefficient of friction ranging from 3 to 10 per cent for those conditions; but with the low coefficient of friction of $\frac{1}{8}$ to $\frac{1}{4}$ per cent for ball bearings a relatively enormous over-stressing of the belt has but comparatively little influence in increasing the journal friction losses.

11 The ball bearing is a most important factor in belt economy, since the absence of the plain bearing friction load permits the use of slack belts and makes for greatly increased belt life. Mr. Fred. Tay-

lor showed the consequent economy most conclusively in his paper, Notes on Belting, (Transactions, Vol. 15, p. 204).

12 *Relative Efficiency of Direct Motor Drive and Ball-bearing Line Shafts.* Mr. Parker refers to the large factor of inefficiency of motors and inquires concerning the possible improvement in line-shaft belt drives due to the use of ball bearings. While in the early days of the introduction of direct-driven tools much was expected from the saving due to cutting out the line-shaft friction, it soon developed that the need for using motors equal to the maximum demand of a tool brought in greater power losses because of such motors working on an average at points of low efficiency.

13 Unless the direct application of the motor results in greater convenience of handling the machine to produce a greater output, the direct drive is not justified. In that case, the mounting of countershaft, loose pulleys and line shaft on ball bearings will result in very considerable power savings. The tests made for the author by Messrs. Dodge & Day on line shafts showed savings of 35 per cent under average conditions; extended to the countershaft and loose pulleys the savings will readily amount to more than half of the total power consumption.

14 In line with this general question Mr. Zimmerman asks whether it would be advantageous to lengthen a group-drive line shaft to 60 ft. to take a larger group involving a shaft length of 100 ft. Unquestionably that will be economical so long as other considerations than those of line shaft and line-shaft motor losses do not govern. As to the relative losses in countershaft and line shaft, it may be said in general that they will be fairly equal. It is true that the countershaft does not run as continuously as does the line shaft, but that simply involves a transfer of the loss from the countershaft hanger to the loose pulleys; only when the belt is actually thrown off does this loss cease; if the loose pulley diameter is decreased, as it should be to decrease the belt tension, the loss is lessened.

15 *Ball vs. Roller Bearing.* Mr. Graves makes inquiry concerning the relative values of ball and roller bearings and their coefficients of friction. The coefficient of friction for good ball bearings has already been given as close to $\frac{1}{8}$ per cent; for roller bearings the friction is about double, assuming always that the rollers are kept in alignment and that hard and true rollers rolling on true and hardened surfaces are used. The real advantage of the ball bearing is not the difference in friction, but its endurance and the consequent permanence of the power saving. As the correct ball bearing employs

only a single row of balls it has no length; that at once cuts out all disturbances, due to deflections of shafts or housings, that seriously affect rollers. The readiness with which the ball bearing is housed to retain its lubricant and to keep out injurious grit, as well as the small space occupied, are also advantages peculiar to it alone. The coefficients of friction cited have been determined by oft-repeated tests. They are referred to the shaft diameter so that the values are directly comparable with those of plain journals.

16 *Reasons for Ball Bearing Cost.* Mr. Parker wishes to know whether the apparently high cost of ball bearings is due to the high cost of manufacture or to large selling expense. Concerning the latter it may be said that the expense of selling ball bearings is not at all high; it is, in fact, lower than in many other lines of high-grade precision machine elements. The cost resides in the absolute necessity for precision, and the character of manufacture. Ball bearings can fitly be compared only with high-grade tools of high-grade steels. The material is a special alloy steel, relatively high in carbon, manganese, chrome and silicon; this is a combination that is very refractory under the cutting tool. After hardening, rough and finish grinding cannot be forced, as that spoils the integrity of the rolling surface. Accuracy of a high degree is essential; the unit of measurement is the ten-thousandth part of an inch. Interchangeability of a high order is not to be secured cheaply.

17 The data showing the saving in power consumption, not in percentage, but in actual consumption, that Mr. Parker asks for, are given in the body of the paper in the table in Par. 36, on lines marked "Plain Bearings measured Kw." and "Ball Bearings measured Kw."

18 *Ball Bearings on Railways.* This use of ball bearings is outside of the subject matter of the paper, but as inquiry has been made by both Mr. Ferris and Mr. Graves it may be noted that ball bearings of the same type are in regular use for main-line railways and electric railways, on the axles in the former and for both axles and motors in the latter. On the Prussian-Hessian State railways the first of these bearings are still in use, and as the result of somewhat over 400,000 kilometers' run (250,000 miles) under standard passenger coaches, show no evidences of wear.

19 In Europe, as well as in the United States, careful comparative measurements, extending over many weeks of 2-min. observations, have shown savings in electric railway power consumption of over ten per cent, with incidental decrease in motor temperature. For main-line and electric railway service the direct power saving is of less

importance than the ability to take advantage of coasting; this saving may frequently rise to 37 per cent. The chief economy lies not in power saving, but in saving of lubricant, attendance, cost of renewals and, in electric railway operation, the keeping of the equipment more in service, and less in the repair shop for renewing bearing linings and rewinding armatures that worn plain bearings have allowed to sag into the polepieces.

20 *Type of Bearing Under Discussion.* The author purposely confined the paper to a report of results of tests made for him

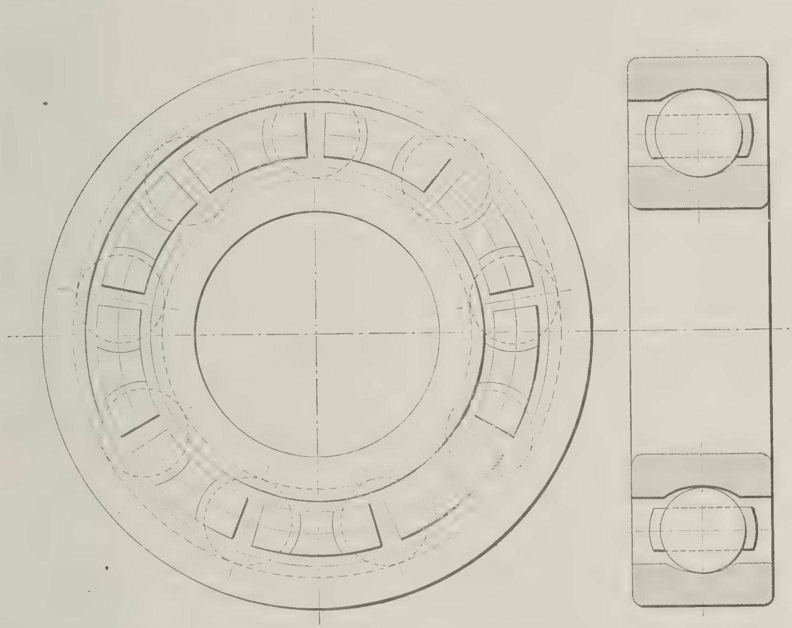


Fig. 1 Elevation and Cross Section of the Hess-Bright Ball Bearing

by Messrs. Dodge & Day, preferring to bring out the engineering value and economic value to be expected of correctly made, correctly selected and properly mounted ball bearings. As Mr. Miller has asked for information concerning the specific ball bearing involved in the test it is proper to say that it is known in the United States as the Hess-Bright or DWF, and in Europe generally as the DWF.

21 Fig. 1 illustrates the ball bearing proper, in cross section and in elevation. It will be seen to consist of an inner race, an outer race, a series of balls, all of special steels hardened throughout, and a cage or

separator for the balls. The ball tracks have curvatures approximating the ball outline, the inner track very closely, the outer track slightly less so. The contact between balls and tracks is on a plane at right angles to the axis of the shaft, thus providing only one point of contact of the ball with each track. The sides of the races are continuous, with no interruption at any point for filling in the balls; that ensures absolutely smooth rolling of the balls and the absence of any possible contact with the edges of filling openings. In lieu of side interruptions or filling openings for the balls, assembly is by eccentric displacement of the two races, filling in balls through the wider space at one side, bringing the races into concentric relation, spreading the balls evenly and retaining them in proper position by the separator.

22 As to the refinement necessary in the making of these bearings, to which Mr. Miller kindly refers from his own observations, I would say that balls must be true to shape and to size within a limit of 0.0002 in. The bearing bore is held within a tolerance of 0.0002 in. +, and 0.0004 in. −. The outside diameter is held within 0.0006 in. +, and 0.0012 in. −, according to size. The width is held within 0.02 in. −. Each finished bearing is gaged for trueness of rotation with reference to the bore, and for trueness of the outer race on the ball circle. Each race is tested for uniformity of hardness, referred to a standard, at four points on each side, or eight per race; the sclerescope is used for this purpose, and that in turn is occasionally checked by the Brinnell, as well as the Turner and the Howe hardness test apparatus.

23 Lest it may appear that these refinements are not necessary, it may be well to say that the knowledge of their necessity has been acquired at great cost; also that only to the most painstaking care in material, treatment and workmanship is the success of the ball bearing due as an every-day reliable element of mechanism. A knowledge of proper proportions for various conditions of load, speed, shock, etc., is, of course, also essential.

THE BEST FORM OF LONGITUDINAL JOINT FOR BOILERS

BY F. W. DEAN, PUBLISHED IN THE JOURNAL FOR OCTOBER 1909

ABSTRACT OF PAPER

This paper deals specially with the defects of the usual form of butt joint used on the longitudinal seams of boilers, in which the inside strap is wider than the outside strap. It gives some history of the joint and discusses some of its defects and suggests a substitution for this form.

While stating that there has never been an explosion of a horizontal return tubular boiler built with the ordinary form of butt joint, the author gives an example of the rupture of such a joint that would have resulted in an explosion. The joint recommended as a substitute for this, is one that has the inside and outside straps of the same width, but the outer row of rivets is made with a wide pitch and the straps are made sufficiently thick to stand caulking between the widely pitched rivets.

Ordinarily the efficiency of this substitute joint is from 84 to 85 per cent, but it can be made as great as that of any other form of joint, if the pitch of the rivets is wide enough, in which case the straps would have to be thicker than would otherwise be necessary.

DISCUSSION

REGINALD P. BOLTON. The form of longitudional joint for boilers, which Mr. Dean has described as the best, is as old as the time of Brunel, and was tested by him in 1838, and again by Longridge in 1857. It is a double-welt triple-riveted joint, omitting alternate rivets in the outer strip, and it has the defect of undue distance for calking between the outer rivets. It is not so good a joint as it would be when the triple riveting is continued, instead of omitting the alternate outer rivet. The other form of joint to which Mr. Dean refers, in which the inside welt was wider than the outside welt, has stood the test of many years usage, and I do not know of any case of failure.

2 In discussing the longitudinal joint, we should not lose sight of the fact that the weak parts of every longitudinal joint are the ends, where the two shell plates unite and the circular seams meet the longitudinal joint. It is there that weakness develops in all joint construc-

tion. In explosion cases on which I have been engaged, I have found that trouble has developed at those points, and have noted that ruptures commenced there. Therefore, in dealing with the design of longitudinal joints, the essential feature seems to me to be its character where it meets the circumferential seam.

E. D. Meier. I think that the value of this joint depends largely on the diameter of the boiler that one has in mind. In a Scotch marine boiler, from 12 to 15 ft. in diameter, the joint would be an excellent one, especially with the scalloped edges mentioned by Mr. Dean. That is a very troublesome thing to do, but in addition to the advantage of the scalloped edge which Mr. Dean cited, there is the further one, that it modifies the tendency, common to such joints, to buckle at the point where the sheets come together. The butt joint is stiffer there than any other part of the shell and with a change in the pressure and temperature the buckling ultimately tends to impair the joint.

2 With a small boiler, 36 in., 42 in., or 48 in., in diameter, the joint is too large a proportion of the total circumference, and this action would become worse. That buckling action is distributed by making the butt plates as thin as possible, and making the inside one longer than the outside one.

3 The one joint that was not considered in the paper—the welded joint—will be an ideal one when we can be sure of a weld that will give 95 per cent efficiency. The difficulty will be to test it. We do know, however, that when we rivet a joint and do it honestly, we have something that can be relied on. Much will depend on how the material is chosen and how the work of laying up and riveting is done. The joint should be made by carefully bending the butt straps at a red heat to the true curve, and rolling the plate itself true to template. This will make as perfect a joint as possible. For a large diameter of boiler, I think the joint advocated by Mr. Dean, especially if the edges are scalloped, is an excellent one, but for smaller diameters I prefer the old joint.

4 Two other points must be considered: first, how the caulking is done, as in many sheets the initial fracture is caused by bad caulking; second, what sort of metal was used, for unless the chemical analysis of the plates as to minimum of injurious metalloids is firmly insisted on, trouble is sure to follow even in the best proportioned joints.

Prof. A. M. Greene, Jr. Mr. Dean is probably aware that in the 1893 report of the Chief of the Bureau of Steam Engineering of the

Navy, it is shown that the boilers intended for the New York, the Columbia and the Minneapolis, were all designed on the same plan as that which Mr. Dean recommends. The illustration in the paper is almost exactly similar to those in the report. These boilers were all installed and have given entire satisfaction.

2 Locomotive engineers, however, are using the unequal length butt strap quite extensively. I know of locomotives in which two rows of rivets were placed outside of the outer butt strap, and I do not know of any failure of such joints. If it is a case of getting increased efficiency, and still having the outer butt strap arranged for a caulking distance, I do not see why we should depart from the method of unequal straps to use the equal strap arrangement which cannot give such high efficiencies.

William A. Jones. I wish to point out the tension which exists in the outer row of rivets and its effect on the drum shell. This should have an important part in determining whether the form of joint which Mr. Dean recommends is really better than if the outer butt strap were cut back one row of rivets on each side, so that the rivets at their caulking edges would be close together.

2 We probably all agree that rivets are more reliable in shear than they are in tension; that the closer and more firmly the edge of the outer butt strap is held down, the less caulking will be required and so the less possibility there will be of injuring the shell plates by caulking the butt strap in the shop, and the more remote will be the probability of subsequent leaks, prompting inexpert men to caulk them again later.

3 If we assume that the inner rivets are about 3 in. apart, then the outer rivets shown in the joint which Mr. Dean recommends will be about 6 in. apart, and each rivet will be holding an area of butt strap of from 15 to 20 sq. in., which, at 200 lb. pressure, will require from 3000 to 4000 lb. tension per rivet. In addition, each of these rivets will be required to hold the caulking for an edge about 6 in. long, and the caulking will have an advantage over the rivet of about 2 to 1, due to the leverage which it has because the rivets are back from the edge. It does not require much thought to see that these rivets would be better able to do this work if they were twice as close together.

4 The joint which Mr. Dean has shown has five rivets in double shear on each side, in a length equal to the pitch of the outer rivets, so that ten times the area of one rivet is the total area in shear in this length. If, on the other hand, the outer butt strap were cut back so

that the rivets at its edge would be close together and the outer rivets were in single shear, then the total area in shear would be only one-tenth less, and the proportion of the circular tension transmitted by the rivets in single shear could not be more than 11 per cent of the total in this case.

5 I understand that it is in an effort to improve the action of this 11 per cent of the force involved that this wide outer butt strap is recommended, and that where four rows of rivets are used instead of six, this proportion may rise to 20 per cent. In any case, the slight in the shell plate is less, I believe, than the bending tendency which the tension would produce in the rivets, due to pressure on the wide outer butt strap.

6 Let us consider the forces acting upon a rectangular area of plate in a drum shell under pressure. The circular tensions acting tangentially at the edges of this area are equal in intensity, but act at an angle to each other, so that each has a component normal to the chord of the area considered. These normal components exactly balance the pressure acting on that chord. When the area considered embraces a half-circle, the normal components become equal to the circular tension.

7 In the case of the outer butt strap, if all the circular tensions of the drum could be transmitted to the outer butt strap by rivets at its extreme edge, the shear of these rivets alone would hold the outer butt strap to the drum, and the components of the shears normal to the chord would just balance the steam pressure on that chord, so that no tension in the rivets would be necessary, except for caulking. Moving the rivets back from the edge of the butt strap makes the shear act more nearly parallel to the chord, while it does not diminish the chord, so that shear alone will no longer hold the butt strap in place, and tension must be developed in the rivets to make up the difference.

8 Transmitting part of the circular tension through the inside butt strap further increases the tension on the rivets, due to pressure, but the additional tension in this case maintains the curve in the inner butt strap by stitching it to the surface which receives the pressure and the reaction of the tension at the inner ends of these rivets is thus provided for.

9 In the case of the outer rivets of the joint which Mr. Dean shows, reaction of this tension at the inner ends of the rivets must be absorbed by an abrupt change in direction of the circular tension at those points, tending to produce corners in the drum shell in order to satisfy the triangle of the three forces formed by the tension on the rivet, the

tangential tension to the right, and the tangential tension to the left. If we assume a 42 in. drum, 200 lb. steam pressure, 6 in. pitch of outer rivets, each of which takes in tension the pressure of 20 sq. in., we have 4000 lb. tension in each rivet due to steam pressure, the inner ends of the rivets being anchored by an abrupt change in direction of about 9 deg. of 25,200 lb. circular tension.

10 Evidently, this abrupt change of direction of the total circular tension may readily distress the plate more in the form of joint which Mr. Dean recommends than in the usual form of joint with the narrow outer butt strap, even though a very small part of the circular tension is transmitted through a rivet in single shear.

11 Mr. Dean's statement that he believes there has been no case of failure of butt strap joints, would indicate that there was nothing wrong with the established form using the narrow outer butt strap. Certainly the remedy proposed seems more objectionable than a rivet in single shear.

SHERWOOD F. JETER.[1] It seems that all engineers design joints with reference to their weakest point, that is, provided the joint was to be ruptured in a machine. Of all explosions that to my knowledge have been due to ruptures, none of them have occurred in the theoretically weakest part of the joint. Most explosions due to rupture of the sheet have occurred near the joint and were apparently due to flexure of the metal, which had destroyed its life at the particular point of rupture.

2 I believe that there is a great need for an investigation as to what causes the rupture of the plate, and for other than machine tests of different kinds of joints. An account in Power states that there have been four ruptures of butt-strap joints of a nature similar to what was previously alluded to as a "lap cracking" of the joint. From the great number of lap joints in successful use for twenty-five years or more, it may be judged that something besides a mere lapping of the plates causes such defects.

THE AUTHOR. There is very little for me to say in closing, as my views have been fully set forth in the paper. I am interested in the history of this joint as stated by Mr. Bolton. I first knew of it in 1889; it is shown in Thomas W. Traill's book on boilers, and a table of sizes of parts is there given.

[1] The Bigelow Co., New Haven, Conn.

2 Several of the speakers express doubt as to the tightness of the joint on account of the wide spacing of the outer row of rivets. There should be no doubt of this kind, for too many of them are in use. I know of one joint with $1\frac{1}{4}$-in. rivets in 1-in. straps on a pitch of $9\frac{1}{8}$ in., and another with $1\frac{3}{16}$ in. rivets in a $\frac{7}{8}$-in. strap on a pitch of $8\frac{3}{8}$ in.

A REPORT ON CAST IRON TEST BARS

By A. F. Nagle, Published in The Journal for Mid-October 1909

ABSTRACT OF PAPER

This paper is designed to show engineers that test pieces, whether cast in separate molds or in the same mold as the main casting, are not *perfect* indications of the character of the iron in the main casting. In other words, uniformity of results is not found in practice where we know of no reason why they should not be uniform. These test bars were used in the construction of over 3,000,000 lb. of pumping-engine castings, involving soft and hard irons for the various parts. Tables 5, 6 and 7 would indicate a probable variation of 15 per cent where uniformity might be expected.

DISCUSSION

Prof. W. B. Gregory. The writer has recently made a large number of tests of cast-iron specimens of one-inch square cross section and with supports 12 in. apart, a few being also broken in tension. The results confirm the deductions of the author as to the relationship between breaking loads in tension and in cross bending. The ten-to-one ratio holds in these tests as in those given by the author. Table 1 gives the results of the cross-bending tests, the load being applied at the center.

TABLE 1 TESTS IN CROSS BENDING

Specimens 1 in. by 1 in., 12 in. between centers, load applied at Center.

Number	Breaking Load Lb. per Sq. In.	Deflection In.
1	2280	0.10
2	2250	0.10
3	2680	0.09
4	2410	0.09
5	2250	0.08
6	2370	0.09
7	2240	0.09
8	2310	0.08
9	2250	0.09
10	2470	0.08
11	2180	0.10
Mean	2335	0.09

2 From the specimens broken in cross bending, six were selected from which were turned tension test pieces approximately $\frac{1}{2}$ in. in diameter at the smallest section, their length over all being 5 in. The threads at the ends were $\frac{3}{4}$ in. outside diameter. The test pieces were made to fit loosely into the tension bars of the testing machine so that side stresses were entirely eliminated, and the specimens were broken in pure tension. The results are given in Table 2.

TABLE 2 TENSION TESTS

Number	Breaking Load Lb. per Sq. In.
1	22900
2	23300
3	22800
4	23550
5	24600
6	22050
Mean	23200

The ratio of tensile strength to load in cross bending is

$$\frac{23200}{2335} = 9.94$$

This comparison can be made only on the basis of averages, as no record was kept of the numbers of the specimens broken in cross bending. The six tension specimens therefore represent six of the eleven specimens broken in cross bending. Specimen No. 9 of the cross-bending tests may be taken as fairly typical of the others. A chemical analysis was made of this specimen with the following results:

Total carbon	4.04
Silicon	1.76
Phosphorus	0.562

3 The mean deflection as given by the author averaged 0.45 in. for two sets of specimens and 0.44 in. for another set. The highest value of deflection in any case was 0.50 in. Since the deflection varies as the cube of the length of specimens between supports, it follows that the deflection for specimens tested with supports 24 in. apart should be eight times the deflection for a length between supports of 12 in. On this basis the specimens tested by the writer should have

$$\frac{0.45}{8} = 0.056 \text{ in.}$$

deflection instead of 0.09 in. average as the tests showed. Can this discrepancy be explained by the difference in chemical composition or is it due to other causes?

4 This raises the question of what deflection ought to be specified for one-inch square specimens with 12 in. between supports. Some specifications have recently been brought to the attention of the writer in which the minimum deflection was placed at 0.15 in. Is this commercial cast iron or does it call for a special mixture, expensive and hard to obtain?

5 The author has mentioned that the "skin of the metal" was of no appreciable thickness. I would like to ask if he has ever tried the effect of rattling on specimens. The process of rattling will remove the sand and the skin of the metal. In this connection the results in Table 3 may be of interest.

TABLE 3 TESTS OF CAST IRON IN CROSS BENDING

SPECIMENS ROUND, $1\frac{1}{4}$ IN. IN DIAMETER, 12 IN. BETWEEN CENTERS. NOT RATTLED.

No.	Breaking Load Lb.	Deflection In.	Remarks
1	2450	0.075	Cast in pairs on end
2	3010	0.08	" " " " "
3	2670	0.07	" " " " "
4	2580	0.14	" " " " "
5	2700	0.09	" " " " "
6	2580	0.14	" " " " "
7	2620	0.08	" " " " "
8	2430	0.075	Cast flat
9	3360	0.09	" "
10	2750	0.08	" "
11	2990	0.09	" "
12	3170	0.09	" "
13	2950	0.095	" "
14	2960	0.12	" "
15	3080	0.10	" "
16	2580	0.075	Cast on end
Mean	2805	0.093	

6 The tests given in Table 4 are on specimens of the same size as those in Table 3. The metal used was as nearly the same as the foundry could make it and the specimens were placed in a rattler and the sand and "skin" removed by abrasion. From these figures it will be seen that rattling has increased the strength, of the specimens, the increase being 3474 − 2805 = 666 which divided by 2805, gives 23.85 per cent. This phenomenon has, been noticed by other experimenters.

TABLE 4

No.	Breaking Load Pounds	Deflection Inches
1	3750	0.09
2	3330	0.095
3	3400	0.08
4	3520	0.09
5	3640	0.09
6	3640	0.10
7	2760	0.075
8	3670	0.095
9	3060	0.09
10	4020	0.10
11	3440	0.09
Mean	3474	0.0904

7 The statement that rattling increases the strength by about 25 per cent seems to be borne out by experiments. The increased strength is probably due to a removal of some of the internal stresses in the specimens and to the fact that the particles of iron, by the process of tumbling the bars together, are allowed to arrange themselves so that they are better able to resist stresses than they were before rattling.

8 Since the breaking load varies directly as the moment of inertia of the cross section of the specimen about the gravity axis, we have

Ig for the specimens $1\frac{1}{4}$ in. diameter $= \frac{1}{4}\pi\gamma^4 = 0.7854 \times 0.625^4 = 0.12$
Ig for the specimens 1 in. square $= \frac{1}{12}bh^3 = \frac{1}{12} = 0.0833$

Then

$$\frac{0.1203}{0.0833} = 1.44$$

Making the comparison between the unrattled round specimens and the square ones, we have

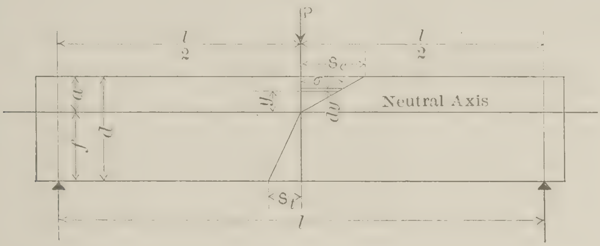
$$\frac{2805}{2335} = 1.2$$

Comparing the rattled round specimens with the square ones we have

$$\frac{3474}{2335} = 1.487.$$

GEO. M. PEEK. The paper brings up a point which I have had in mind for some time, and which I have never seen explained in any of the text books on the strength of materials or any of the engineer's

hand books. The formula in Par. 15 is obviously not applicable to cast-iron beams, for the reason that it assumes that the neutral axis of a rectangular beam is in the center, which is true only when the beam is made of a material with equal tensile and compressive strengths.



2 In order that we may be able to construct a formula to be used in the design of a beam made of material whose crushing and tensile strengths are not equal, we must know the ratio between them. It may be reduced as follows, referring to Fig. 1, herewith:

Let M = bending moment = $\frac{Pl}{4}$ for load at center of span.

P = load at center.
l = length between supports.
S_c = compressive strength.
S_t = tensile strength.
b = breadth of beam.
d = depth of beam.
a = distance to extreme fiber on compression side.
f = distance to extreme fiber on tension side.
K = ratio compressive strength to tensile strength.

All dimensions are in inches.

3 We have the moment of resistance on the compression side

$$\int_0^a b\sigma y\,dy = \int_0^a bS_c\frac{y^2}{a}dy = \frac{ba^2}{3}S_c$$

and in like manner we find the moment of resistance on the tension side to be $\frac{bf^2}{3}S_t$. Since these two resistances are on opposite sides of the neutral axis they must be equal, or

$$\frac{b\,a^2}{3}S_c = \frac{bf^2}{3}S_t \text{ or } \frac{a^2}{f^2} = \frac{S_t}{S_c} = \frac{1}{K} \therefore f = a\sqrt{K}$$

4 Since the sum of these two resistances must be equal to the bending moment we have

$$M = \frac{b\,a^2}{3} S_c + \frac{bf^2}{3} S_t$$

Substituting KS_t or S_c

$$M = \frac{b\,S_t}{3} (K\,a^2 + f^2) = \frac{P\,l}{4}$$

$$P\,l = \frac{4}{3}\,b\,S_t\,(K\,a^2 + f^2) = \frac{8}{3}\,b\,S_t\,a^2\,K$$

$$d = a + f = a\,(1 + \sqrt{K})$$

$$\therefore\ a^2 = \frac{d^2}{(1 + \sqrt{K})^2}$$

Substituting again

$$P\,l = \frac{8}{3}\,b\,d^2\,\frac{K}{(1 + \sqrt{K})^2}\,S_t$$

$$S_t = \frac{3}{8}\,\frac{(1 + \sqrt{K})^2}{8\,K}\,\frac{P\,l}{b\,d^2}$$

If we substitute 1.747 for K we get $S_t = \frac{P\,l}{1.155\,b\,d^2}$ or Clark's formula.

5 Taking the average compressive strength of cast iron as 112,000 lb. per sq. in., and the average tensile strength as 28,000, or $K = 4$, we have

$$S_t = \frac{P\,l}{1.185\,b\,d^2}$$

6 Applying this formula as Mr. Nagle does Clark's, in Par. 15, we have

$$\frac{2372 \times 1.185 \times 2 \times 1 \times 1}{24} = 2350$$

As will be seen, this formula gives results within 1.4 per cent of those obtained from the test.

A. A. Cary. It is unfortunate that the value of the structural study of metals and alloys, by use of the pyrometer and microscope, is

not more widely appreciated. I feel safe in saying that by such means all variations such as noted in Mr. Nagle's paper can be most satisfactorily accounted for. In iron and steel the fact is now generally recognized that metals identical in chemical composition may possess widely differing mechanical properties which are quickly recognized by microscopic examination.

2 Chemical analyses, as given in Table 1 of the paper, are undoubtedly of considerable value in the investigation of cast-iron; but without a physical examination our knowledge of the ability of the metal to withstand stresses and strains is very uncertain. Not only will investigations of this kind show us the cause of the variations noted in Mr. Nagle's paper, but they will give us the information needed to produce a metal of great uniformity.

Prof. T. M. Phetteplace. It would be interesting to know whether a thorough sand-blasting would have any effect, as different results seemed to be obtained by cleaning off the skin of the material

The Author. Since the paper was written I have had opportunity to examine some instructive records of eleven sets (of three each) of round test bars. The bars were $1\frac{1}{4}$ in. in diameter, rough, on 12-in. supports, the breaking loads being corrected for actual diameters. The deflections were not corrected.

BREAKING LOADS IN POUNDS. DEFLECTION FROM 0.12 IN. TO 0.15 IN.

1........	3276	3185	3044	4400	4005	2913	3276	3306	3382	3204	3268
2........	3367	3276	3162	3100	3913	3003	3185	3204	2976	3204	3124
3........	3276	3534	3255	3500	3640	3115	3026	2937	3003	2912	2812

2 The three bars in each set were cast in three separate molds, No. 1, or the upper line, being cast from the first pour of the ladle, No. 2 from the middle and No. 3 from the bottom. It will be observed that in eight of these eleven sets, the bar selected from the two nearest in agreement, came from the middle of the pour, and that all of the extreme variations were found in either the first or last pour. If we have only two bars they would differ as much as 22 per cent, while if we took the two out of three nearest in agreement, those two would not vary more than 2 per cent or 3 per cent.

3 I am very glad that Mr. Peek has taken up the mathematical solution of fitting a formula to the facts. Whether his demonstration or Clark's is the correct, or the better, one, I will not attempt to say,

but it is a pleasure to find that the two methods agree so well with the facts. I trust that this publicity will banish the old form of formula from the text books.

4 Professor Gregory has made an oversight in the dimensions of my bars. Being twice as wide as his, the deflections do not show very great variations: as $4 \times 0.09 = 0.36$ to 0.45, instead of $8 \times 0.09 = 0.72$ to 0.45.

5 I have had no experience with bars 1 in. by 1 in. by 12 in., but I think that 0.15 in. deflection would be difficult to realize in machinery castings.

THE BUCYRUS LOCOMOTIVE PILE DRIVER

By Walter Ferris, Published in The Journal for November 1909

ABSTRACT OF PAPER

This paper describes a new railway pile driver recently put on the market. The leading feature is a very powerful propelling apparatus and a large boiler, enabling it to act as a locomotive and haul its own train of tool cars, boarding cars, etc., over the road.

In order to transmit more than 250 h.p. to the axles of ordinary bogie trucks, which do not remain in line with the car body when passing curves, a special type of driving connection has been developed and is described in detail with drawings. The machine carries the pile driver apparatus at one end or the car, with power devices for raising the leaders and for swinging them to either side of the track as desired.

To enable the machine to drive piles at the other end when no railway turntable is at hand, a special turntable is attached to the under-side of the main car sills just above the track. This consists of hydraulic lifting apparatus and a large ball bearing upon which the entire pile driver, including trucks, is lifted clear of the track and turned end for end in from ten to fifteen minutes.

DISCUSSION

A. F. Robinson.[1] I feel very much pleased with the behavior of this driver as far as we have gone. I am especially pleased with the last three drivers, which are equipped with the extra high-speed gear. Our men find in handling this driver that it saves a good deal of time over the locomotive, especially in the short moves required in spotting the pile for driving and also the short run back to the end of a bridge to obtain piles.

2 As soon as this machine is thoroughly understood a great many will be used. This will be especially the case when we use reinforced-concrete piling more extensively.

L. J. Hotchkiss.[2] There are in use many antiquated pile drivers which are slow and difficult to handle. In some cases the leaders

[1] Bridge Engineer, Atchison, Topeka and Santa Fé Ry.

[2] Asst. Bridge Engineer, Chicago, Burlington & Quincy R. R., Chicago, Ill.

must be raised by means of a set of blocks attached to the track ahead of the driver, the fall line being carried to a spool on the engine. With such a machine ten minutes may be required to raise or lower the leaders. Where the work is not too far from the station, and there are no overhead obstructions, it may not be necessary to lower the leaders when running to the station. In many places, however, the leaders must be lowered every time the pile driver goes in, and raised again on coming out. On a busy single-track railroad this may cause much loss of time in the course of the day.

2 The time loss may not be merely that directly caused by slow handling of the machine. In many locations the movement of trains is such that there are several periods during the day when with a quickly operated driver there is just time between trains to run out, drive one or two piles and get in the clear again. With a driver operated as previously described this cannot be done, as so much time is required to handle the leaders that there is none left for driving piles. There are, however, drivers which do not have this objection but which must be handled by a locomotive. This is expensive in two ways. There is charged to the work of pile driving the cost of engine service, and the locomotive is kept out of regular train service. In times of heavy business the latter item is in itself one of considerable importance.

3 The self-propelling feature of the machine described by Mr. Ferris, its large boiler capacity and the arrangement for turning it, are its most prominent features. As stated by Mr. Ferris, the usual charge for a locomotive and crew is from \$20 to \$30 per day, \$25 being assumed as a fair average charge. The locomotive will furnish steam for the driver, making a fireman on the latter unnecessary. In the case of the self-propelling driver it is necessary to have a fireman, and as the machine is somewhat complicated, better men must be employed both as engineer and as fireman than would be needed ordinarily. For this reason the net saving by cutting out engine service probably will not exceed \$20 per day. It is not unusual to have from 600 to 800 piles to drive on one division in a single season. If we estimate that 20 piles a day are driven, and this number is well above the average, 30 days will be required to drive 600 piles. For this period the charge for engine service would amount to \$600, which is 5 per cent on an investment of \$12,000. It will thus be seen that the elimination of engine service in pile-driving work is a matter of no small importance.

4 A machine such as Mr. Ferris describes has sufficient power and

steaming capacity to handle its own train a considerable distance. Where a long haul is to be made the propelling mechanism is quickly thrown out of gear and the whole outfit put in a regular train. One of these pile drivers recently handled a train consisting of four bunk cars, a locomotive tender fully loaded with coal and water, one car containing 40 tons of coal, and a way car. This train was taken up a 1.4 per cent grade more than a mile long. A few days later this driver hauled 140 tons in addition to its own weight up the same hill at about 7 miles per hour. The steam gage showed 175 lb. pressure when the top of the hill was reached.

5 The conditions of railroad operation today require that all possible economies be made both in operation and construction. The locomotive pile driver of large capacity is a recent development and one which must still be regarded, to a certain extent, as an experiment. Experience so far, however, indicates that it is an economical machine, in that it dispenses with locomotive service and is quickly handled on all classes of work.

The Author. The railway pile driver is used for two general classes of work, construction and maintenance. For construction work, in most cases, almost any track machine which is capable of driving piles will answer the purpose fairly well, because in such work the machine, if fairly well fixed, is able to stand for considerable periods of time at one place, and efficiency as a pile driver is the leading object.

2 In maintenance work, however, which generally consists in repairs, such as strengthening the abutment of a bridge which is showing some signs of washing down, or especially in repairs after a washout, the mobility of the machine is the leading feature. To illustrate this point, I may say that the first machine of this design which we built was tried out at a bridge in California which was a mile and a half from the nearest railroad siding. I happened to be with that machine at the time, and during the forenoon we ran it out from the siding to the bridge we were repairing, and back into the siding again, seven times, to dodge passing trains. During this time twelve piles were driven, one or two at each trip.

3 The base price of this machine is $11,650 without the turntable and the steam hammer. As the turntable and steam hammer, and electric light plant and other extras are added, the total price may run to something about $14,000. This represents an increase of cost to the railroad, above what they have been accustomed to pay

for a pile driver, of $3,000 to $4,000 for each machine. The experiment in the case of this machine was quite as much in the line of commercial engineering as of mechanical engineering. When we built the first machine we were a good many thousand dollars behind, and somewhat doubtful if we would get it back. It looks now as if the machine would take very well. The operating department of the Southern Pacific, to which we recently furnished a machine, had previously charged the bridge department $45 a day for the use of a locomotive, which was dispensed with by the use of a machine capable of doing its own propelling work.

THE PITOT TUBE AS A STEAM METER

By George F. Gebhardt, Published in The Journal for Mid-November 1909

ABSTRACT OF PAPER

The application of a pitot tube system along the lines described in the paper is an accurate means of determining the *velocity* of steam at any point in a pipe, provided the values of the various influencing factors are known; and for straight lengths of piping with continuous flow, under these conditions, it is an accurate means of determining the *weight* of steam flowing.

Under average commercial conditions in which the pressure and quality of the steam fluctuate and an average value must be taken for the density of the self-adjusting water column, only approximate results can be obtained, the extent of error varying with the degree of fluctuation.

For velocities in excess of those corresponding to a 1½-in. water column (about 2000 ft. per min. for pressures over 70-lb. gage pressure), tests gave a maximum error of about 2 per cent for continuous flow in straight lengths of piping.

The coefficient of the tubes, as applied in Fig. 12, is practically unity and no calibration of the apparatus is necessary.

Further tests are necessary to show whether application, as in Fig. 13 and Fig. 14, gives reliable results.

DISCUSSION

Prof. W. B. Gregory. The pitot tube was invented in 1730. An account of the tube and the manner in which it was invented may be found in Histoire de l'Academie des Sciences for 1732. This paper by Pitot is of considerable interest. Some of its accompanying drawings are reproduced in a paper which I presented before the Society on The Pitot Tube, published in the Transactions, Vol. XXV. The statement of Mr. Gebhardt that the tube was first used in 1837 is evidently a misprint.

2 The author has apparently developed a practical instrument of real value. However, it seems to the writer that the device for determining aspiration effects can not be relied upon to make determinations of any value. Fig. 6 shows a special fitting, which, after pipes are screwed into the two ends, will be anything but an ideal fitting to

give correct static pressures. Most of the trouble in the past has been on the static side. The fitting shown amounts to an enlargement of the pipe beyond the end of the entering pipe and then a contraction where the steam enters the outgoing pipe. Serious eddying must result and it does not seem at all likely that the slots *ss* are long enough to neutralize the effect of the eddying and the change of section. Even if they do correct these errors it does not follow that *B* is located where it will give the correct mean pressure in the special fitting. The change of section and consequent eddying may change the pressure along the special fitting so that the pressure shown at *B* is not the true mean pressure.

3 I would like to ask Professor Gebhardt if he has used static openings about 1/16 in. in diameter drilled at right angles to the axis of the pipe? Extensive experience with the pitot tube as a device for measuring the velocity of water has taught me to avoid irregularities in a pipe, due to special fittings or other causes, when the static pressure is taken from the walls of the pipe. An unobstructed length of straight pipe is absolutely essential to accurate work.

4 The desirability of finding the correct static pressure is apparent as it seems probable that one constant would apply to reduce velocity at the center to mean velocity, in any and all sizes of pipe. The experimental determination of the correct angles for the static nozzle, as shown in Fig. 5, would then be avoided.

Walter Ferris. The remarks of Professor Gregory in regard to the special fitting for finding the effect of aspiration reminded me forcibly of an experience a few years ago with both a venturi meter and a pitot tube for measuring water. Perhaps the conclusions at which I arrived at that time may be suggestive, although possibly not of direct application in the case of a steam meter.

2 Until quite recently, that is, within a few years, I think it has been assumed that it was necessary, in the use of the pitot tube, to have a static tube close to the dynamic tube, or at least at the same distance from the walls of the conduit. I believe that William Monroe White, six or seven years ago, made some experiments demonstrating that the velocity head taken from the impact side of a pitot tube is correct, whatever the shape of the nozzle, so long as it is a surface of revolution. Thus the nozzle may be either cylindrical, or a converging or diverging cone, and the dynamic head will be correctly indicated, any variations in the coefficient of the pitot instrument as a whole being due to the shape or location of the static opening.

3 In the venturi meter, we find that the static pressure is always taken from the walls of the conduit, where the velocity may not be over half the maximum velocity, and yet the results from the venturi meter are invariably correct to within one per cent, if conditions are favorable. Therefore a dynamic nozzle, which is a surface of revolution, combined with a static nozzle terminating in the wall of the conduit (as in the venturi meter) should together form a pitot instrument which is correct to the formula, and needs no calibration. This seems to indicate that for a pitot instrument to measure the flow of water it is not necessary to take the static head and the dynamic head in regions of the same velocity, and that the true average static pressure will be indicated through intervening velocities, and correctly registered, even when the piezometer is located in a region of low velocity. From this I infer that in this steam meter sufficiently small static openings in the true smooth wall will probably give correct results as they do in the water meter, although I have no experimental data with which to confirm this opinion.

A. R. Dodge. I would like to take exception to a statement in Par. 6: "On account of the great density of mercury and the variation in height of the condensed vapor above the mercury, this application of the pitot tube has very little value scientifically or commercially." The General Electric Company has developed a steam meter, both of the indicating and recording types and has built several hundred of these meters using mercury and condensed vapor above the mercury. This condensed vapor automatically remains at a constant head.

2 Recently three recording meters, selected at random out of a lot of fifty, showed a maximum error of less than two per cent. Ninety per cent of the readings were within one per cent on the three meters, which had an automatic pressure correction and also a temperature correction. These meters can be used on any size of pipe, from 2 in. up to 36 in., the 36-in. pipe, of course, being for atmospheric conditions of steam. These steam meters we have found to be valuable in improving the consumption of steam in our various plants.

3 We have also experimented with several of the types described in this paper in which mercury is not used and have found them excellent in many respects, but the use of mercury is not at all objectionable.

The Author. Prof. W. B. Gregory is correct in his statement concerning the defects in the apparatus for determining aspiration

as illustrated in Fig. 6. This drawing refers to an old discarded fitting and was published through an oversight. The apparatus used in connection with the tests recorded in the paper is the same in principle but differs in the details of construction. The inner surface of the fitting is of the same degree of smoothness as that of the pipe. This inner diameter of the chamber corresponds to that of the pipe. The ends of the pipes are threaded and finished in such a way as to fit snugly against the threaded end of the fitting, forming a practically continuous tube of uniform diameter. The slots are 10 in. long and ¼ in. in thickness. Careful measurements with searching tubes and delicately balanced differential manometer failed to show eddies of appreciable magnitude.

2 Static openings, about one-sixteenth of an inch in diameter, drilled at right angles to the axis of the pipe, showed no aspiration effects at velocities up to 15,000 ft. per min. (the maximum obtained during the tests) but are unsuitable for the appliances described. It is the author's intention to develop a simple meter which can be constructed of standard fittings and which may be attached by tapping the pipe in the ordinary way. Such an application necessitates the projection of the static nozzle beyond the inner surface of the pipe, an arrangement which causes serious aspiration. With a standard ½-in. nipple projecting ½-in. beyond the inner surface of a 3-in. pipe an aspiration effect corresponding to 10 in. of water was noted at a velocity of 12,000 ft. min. (pressure 100 lb. gage). At a velocity of 6000 ft. per min. the aspiration amounted to 1½-in. of water. It was for the purpose of neutralizing this aspiration that the static nozzle was cut at an angle, as indicated in Fig. 5.

3 Mr. Ferris' remarks are in accordance with experiments conducted by the author, but, as stated above, a static opening terminating with the inner wall of the conduit is not applicable to the instruments in question. Fig. 1 illustrates such a static opening, but in the actual construction the nozzle projected ½ in. beyond the inner surface.

4 Mr. Dodge's experiments with the use of mercury as an indicating medium are of considerable interest, in that they show the development of a practicable and accurate steam meter which is little known to the general engineering public. It would be of great interest if Mr. Dodge would describe the instrument used at the works of the General Electric Company and give some of the test results.

GENERAL NOTES

AMERICAN SOCIETY OF CIVIL ENGINEERS

At the meeting of the American Society of Civil Engineers, February 2, two papers were presented for discussion: Underpinning the Cambridge Building, New York City, by T. Kennard Thomson, Mem.Am.Soc.M.E., and Building Agreements, by Wm. B. Bamford. The papers presented at the meeting of February 16 were: The Effect of Alkali on Concrete, by Geo. Gray Anderson; and Precarious Expedients in Engineering Practice, by John Hawkesworth. On March 2, a paper entitled The Improved Water and Sewage Works of Columbus, Ohio, will be presented by John H. Gregory.

AMERICAN INSTITUTE OF MINING ENGINEERS

The Society takes pleasure in announcing the invitation extended to the members of this Society by the American Institute of Mining Engineers, to attend their Convention at Pittsburg, Pa., beginning Tuesday evening, March 1, 1910. The members of this Society will be welcome at the professional sessions and at such of the excursions as may be undertaken, where the number does not exceed the available facilities.

The Institute headquarters will be at the Hotel Schenley, where a bureau of information will be maintained, and the sessions will be held at the Carnegie Library, opposite the hotel. The Secretary of the Local Committee is Harrison W. Craver, Carnegie Library, Pittsburg, to whom should be addressed all inquiries concerning local matters and arrangements of the meetings.

There will be an excursion to the steel plant at Homestead, which will occupy one day, and an afternoon will be devoted to a visit to the testing station of the United States Geological Survey, where special tests will be made showing the effect of various explosives on mine gas, etc., also some tests on reinforced concrete beams. Arrangements will also be made for a visit to a coal mine and to various manu-

facturing plants. Details of the sessions and excursions will be given in the program furnished to each guest on registration at the headquarters.

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

The Society of Naval Architects and Marine Engineers is arranging for representatives to attend the fiftieth anniversary of the founding of the Institution of Naval Architects, to be held in London, July 5, 1910, and to be made the occasion of an international congress.

Papers and subjects connected with naval architecture and marine engineering will be read and discussed and the attendance of a large number of distinguished naval architects, shipbuilders and marine engineers from all parts of the world is anticipated.

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

An invitation to the International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, to be convened at Dusseldorf, June 20–23, 1910, has been extended to the members of The American Society of Mechanical Engineers. This notice is published for the benefit of individual members who may be able to attend, as the Society is unable to accept as a body the invitation to be present.

NATIONAL CIVIC FEDERATION

The following Honorary Vice-Presidents were appointed to represent the Society at the conference of the National Civic Federation in Washington, D. C., January 17–19, 1910: Jesse M. Smith, Past-President, Chas. Kirchhoff, A. W. Burchard, E. G. Spilsbury, F. M. Whyte and Wm. H. Wiley.

The conference was called to consider uniform state legislation and has formed itself into a permanent organization for the purpose with Alton B. Parker as President. Annual conferences will probably be called. The conference endorsed the conservation of American forests and referred the matter of uniform state laws, providing for right methods of forests taxation and for the effective protection of forests from fire, to the Commission on Uniform State Laws. The regulation of water power by state and federal control was also recommended. A number of other resolutions were passed

upon subjects of national importance, urging uniformity in laws relating to taxation, insurance, child labor, public accounting, legal procedure, etc.

NEW YORK ELECTRICAL SOCIETY

On January 27, 1910, Prof. W. S. Franklin of Lehigh University gave a lecture before the New York Electrical Society, 29 West 39th Street, New York, on The Practical Applications of the Gyrostat. Professor Franklin discussed the physical action and the establishment of the kinematical diagram of the gyroscope, the gyrostatic action of the flywheel of the automobile engine and on shipboard, as well as of the boomerang, Schlick's device for the prevention of rolling of ships at sea, and the Brennan monorail car.

ENGINEERS CLUB OF PHILADELPHIA

The thirty-first annual meeting of the Engineers Club of Philadelphia was called to order by the President Dallett, February 5, 1910, with 129 members and visitors in attendance. An address on Recent Developments in Engineering Practice was made by President Dallett. Following a report of the tellers the following were declared elected as officers of the club: Wm. Easby, Jr., president; Chas. Hewitt, vice-president; W. Purves Taylor, secretary; E. J. Kerrick, treasurer.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The regular monthly meeting of the American Institute of Electrical Engineers was held in the auditorium of the Engineering Societies Building, on Friday, February 11, 1910. W. Lee Campbell of the Automatic Electric Company of Chicago presented a paper entitled, A Modern Automatic Telephone Apparatus. A complete installation connected up for service was on exhibition.

At the annual dinner of the Institute, held at the Hotel Astor on Thursday evening, February 24, Dr. Elihu Thomson, to whom has been awarded the first Edison Medal, was the guest of honor. The following were the speakers of the evening: Dr. John H. Finley, president, College of the City of New York, Education and Invention; Samuel Insull, president Edison Medal Association, Meritorious Achievement in Electrical Engineering; Lewis Buckley Stillwell,

president, A.I.E.E., The Edison Medal, with response by Dr. Thomson. Mr. T. C. Martin acted as toastmaster.

WESTERN SOCIETY OF ENGINEERS

The Chanute medals of the Western Society of Engineers, founded by Dr. Octave Chanute, have been awarded for 1908 to Horace E. Horton, Prof. A. N. Talbot and Morgan Brooks, Mem.Am.Soc.M.E. Mr. Horton's paper was Compression Bridge Members, Professor Talbot's a report of Tests of Reinforced Concrete and Cast-Iron Pipe, and Professor Brooks' was Alternators in Parallel.

SHEFFIELD SCIENTIFIC SCHOOL NEW LABORATORY OF MECHANICAL ENGINEERING

A gift of $250,000 has recently been received by the Sheffield Scientific School of Yale University, from George G. and William S. Mason, graduates in the class of 1888, to be expended for the construction and equipment of a new mechanical engineering laboratory, on a site to be provided by the Board of Trustees. The laboratory will be located on Hillhouse Avenue, will be four stories in height, and will contain approximately 50,000 sq. ft. of floor area and 880,000 cu. ft. of space. The entire equipment will be new and will consist of the most modern appliances for assisting the student in studying the fundamental principles of applied science closely related to mechanical engineering, such as the strength of materials, the combustion of fuel in furnaces and in internal-combustion engines, the making of steam in boilers of different types, the using of saturated and superheated steam in engines or steam turbines, the artificial production of cold, the production, transmission and use of compressed air, the pumping of water, the transmission of power, and the problems of heating and ventilation. It is expected that this laboratory will furnish a field for research work in engineering science, as well as undergraduate and graduate instruction. It is expected that the building will be completed and equipped by June 1911.

COLUMBIA UNIVERSITY COURSE IN WORKS MANAGEMENT

A series of twenty lectures by non-resident lecturers is being conducted in the Department of Mechanical Engineering, Columbia

University, constituting a course in Works Management. Lectures are given on Thursdays and Mondays of each week, at 4.10 p.m., beginning February 10 and closing May 14, in Room 301 Engineering. The course consists, in the following order, of six lectures by Charles B. Going, managing editor of the Engineering Magazine; four by Charles U. Carpenter, Mem.Am.Soc.M.E., president of the Herring-Hall-Marvin Safe Company; two by H. L. Gantt, Mem.Am. Soc. M.E., one by Walter M. McFarland, Mem.Am.Soc.M.E., vice-president of the Westinghouse Electric & Mfg. Co.; three by Harrington Emerson, Mem.Am.Soc.M.E.; three by Richard T. Lingley, C.P.A., treasurer of the American Real Estate Company; and a concluding lecture by Edwin J. Prindle, member of the New York Bar.

STEVENS INSTITUTE ALUMNI DINNER

The ninth annual dinner of the Stevens Institute Alumni Association was held in the Hotel Astor on February 12. Among the speakers were Dr. Alex. C. Humphreys, Mem.Am.Soc.M.E., President of the Institute, Dr. H. S. Pritchett of the Carnegie Foundation, and Col. G. B. M. Harvey of Harper's Weekly.

PERSONAL

A. Bement has been elected second vice-president of the Western Society of Engineers. Mr. Bement presented a paper on the Chicago Harbor Problem before this Society at their meeting of February 16.

Morgan Brooks has been awarded one of the Chanute medals of the Western Society of Engineers for 1908. His paper was on Alternators in Parallel.

C. P. Chester, formerly superintendent of the Morenci Water Company, Morenci, Ariz., has opened a consulting engineering office in El Paso, Texas.

C. W. Comstock has been appointed president of the Comstock-Wellman Bronze Company, Cleveland, O.

Thomas F. Cooke has formed a partnership for consulting engineering with Richard L. Webb, under the name of Webb & Cooke, with an office in Buffalo, N. Y. The firm will make a specialty of power costs.

Fred H. Daniels has been decorated with the Cross of Knighthood of the Northern Star by King Gustav of Sweden in token of his work as an engineer and for courtesies extended to Swedish engineers in this country.

Arthur Falkenau, formerly president of the Falkenau-Sinclair Machine Company, Philadelphia, Pa., has become associated with George K. Hooper, New York.

J. Edwin Fulweiler has become associated with the United Gas Improvement Company of Philadelphia. He was until recently in the engineering department of the Otto Gas Works, Philadelphia, Pa.

W. B. Gregory has been elected president of the Louisiana Engineering Society.

Edwin J. Haddock, formerly chief engineer of the chain department of the Jeffrey Manufacturing Company, has given up his office in Columbus, O., to become mechanical and structural engineer of the Tennessee Coal, Iron & R. R. Co., in the coal mining department, with office at Birmingham, Ala.

F. A. Hall, manager of the chain block and hoist department of the Yale & Towne Mfg. Co., New York, has resigned that position to become vice-president and treasurer of the Cameron Engineering Company, of Brooklyn, N. Y.

F. A. Halsey sailed January 20 on the steamship Arabic for a cruise to the Mediterranean and the Orient. Mr. Halsey expects to be gone about three months and before returning intends to visit some of the important industrial centers of Europe.

Walter Laidlaw, formerly vice-president and general manager of the Snow Steam Pump Works, Buffalo, N. Y., has become identified with the International Steam Pump Company, New York.

Wm. Y. Lewis, formerly manager of the erecting department of the International Steam Pump Company, has established an office of his own at 49, Queen Victoria St., London, E. C., as advisory engineer.

W. A. McFarland, for many years superintendent of the Washington, D. C., water works, has opened an office in the Washington Loan and Trust building, as consulting engineer in matters relating to water works and power plants. He will be associated in a consulting capacity with the engineering firm of Beale & Meigs, which carries on a general engineering practice.

C. J. Morrison, until recently connected with the Emerson Company, has opened an office in New York for efficiency engineering work.

Leslie Moulthrop has received his discharge from the Superior Court as receiver of the Dwight Slate Machine Company, Hartford, Conn., having paid the general creditors in full. The company will be conducted under the same name by a new organization.

Albert Spies has retired from the editorship of the Electrical Record to become the managing director of Foundry News, a new illustrated monthly publication devoted to the foundry arts.

George F. Starbuck, formerly draftsman of the mechanical department of the N. Y., N. H. & H. R. R., New Haven, Conn., has become associated with the Boston Elevated Railway, Boston, Mass., as draftsman in the department of rolling stock and shops.

Cecil H. Taylor has been appointed chief engineer of the Hudson Motor Car Company, Detroit, Mich. He was formerly designing engineer of the Chalmers Motor Car Company, Detroit, Mich.

Charles E. Waddell, formerly consulting engineer, Biltmore Estate, Biltmore, N. C., has established offices for general engineering practice in Asheville, N. C.

Gilbert S. Walker, formerly located at Wheeling, W. Va., has become connected with the Isthmian Canal Commission, Washington, D. C.

James T. Wallis, superintendent of motive power on the Erie division of the Pennsylvania R. R. and the Northern Central has been appointed acting superintendent of the West Jersey & Seashore, R. R., also of the Philadelphia and Camden Ferry, with office at Camden, N. J.

C. H. Zehnder has been elected vice-president of the Empire Steel & Iron Co.

W. H. Zimmerman, formerly manager of the Michigan Power Company, Lansing, Mich., has been retained by the Michigan Railroad Commission as consulting engineer.

CURRENT BOOKS

Engineers' and Firemen's License Law; Boiler Inspection Law; Rules Formulated by the Board of Boiler Rules. Pamphlet issued by the Commonwealth of Massachusetts, 1909. *Wright & Potter Printing Co., Boston, Mass., 1909.* Viii + 67 pp., illustrated.

Contents: Engineers' and Firemen's License Law; Boiler Inspection Law; Rules formulated by the Board of Boiler Rules; Recommendations made by the Board of Boiler Rules; Index to Rules.

Fowler's Electrical Engineer's Pocket Book. Edited by Wm. H. Fowler. 10th annual edition. *Scientific Pub. Co., Manchester, England, 1910.* Cloth, pocket book size, 575 pp., illustrated. Price 1/6 net.

Contents: Miscellaneous Tables, etc.; Wire Tables; Magnetism and Magnetic Data; Conductors and Insulating Materials; Electric Lighting and Wiring; Comparison and Measurement of Resistances; Electrical Measuring Instruments; Electricity of Meters; Primary and Secondary Batteries; Dynamos and Motors; Alternate Electric Currents; Alternators; Transformers; Alternate Current Motors; Switch boards, Circuit Breakers and Lightning Arresters; Electrical Power Transmission and Distribution; Rotary Converters; Electric Traction; Rules and Regulations.

Fowler's Mechanical Engineer's Pocket Book. Edited by Wm. H. Fowler. 12th annual edition. *Scientific Pub. Co., Manchester, England, 1910.* Cloth, pocket book size, 653 pp., illustrated. Price 1/6 net.

Contents: Miscellaneous Tables and Formulæ; Steam Boilers and Fittings; Fuels and Combustion Steam Engines; Steam Turbines; Locomotives; Steam Tables; Valves and Valve Gear; Gas Engines; Gases used in Gas Engines; Oil Engines; Hydraulics; Pumps and Pumping Arrangements; Gearing and Lubrication; Hoisting and Lifting Machinery; Mining Machinery and Appliances; Metallurgy of Iron and Steel; Strength of Metals and Alloys; Beams and Pillars; Springs; Chemistry; Ventilation and Heating.

Slide Rule. An Elementary Treatise. By J. J. Clark,M..E. *Technical Supply Co., New York, 1909.* Cloth, 6 vo., 62 pp., with diagrams. Price, 45 cents.

Contents: Introduction; the Mannheim Slide Rule.

Smoley's Tables. Parallel Tables of Logarithms and Squares, Angles and Logarithmic Functions, with complete set of Five-Decimal Logarithmic-Trigonometric Tables. By Constantine Smoley, C.E. 5th edition, revised. *Engineering News Pub. Co., New York, 1908.* Morocco, pocket book size. Price, $3.50.

Contents: Parallel Tables of Logs and Squares; Table of Bevels; Multiplication Table; Explanation and Examples; Constants; Decimal Equivalent; Logarithms of Numbers; Log. Functions by 10″; Angles Between 0° and 1° Log. Functions by 1′; Natural Functions; Formulæ; Constants: Decimal Equivalents.

TIME AND ITS MEASUREMENT. Ry James Arthur. Reprinted from Popular Mechanics Magazine, Chicago, 1909. Cloth, 12 vol., 64 pp., illustrated.

Contents: Historic Outline; Japanese Clocks; Modern Clocks; Astronomical Foundation of Time.

FOWLER'S MECHANICS' AND MACHINISTS' POCKET BOOK AND DIARY, 1910. Edited by Wm. H. Fowler. 2d edition. *Scientific Pub. Co., Manchester, England, 1910.* Cloth, pocket book size, 448 pp. Price 6d.

Contents: Handy References and Tables; Mensuration, Geometry, and Trigonometry; Uses of Logarithms and Antilogarithms; Materials Used in Machine Construction; Machine Tool Design; Proportions of Machine Tool Parts; Metal Cutting Tools; High Speed Tool Steels; Drilling and Boring Metal; Screw Threads, Screw Cutting, and Taper Turning; Emery and Emery Wheels; Shop Practice; Wheel Gearing; Belt and Rope Driving, Shafting; Lifting Ropes and Chains.

KEMPTHORNE'S RAILWAY STORES PRICE BOOK. Being a Handbook of Prices of Stores and Material used in the Construction and Maintenance of Railways. By William Oke Kempthorne. *E.&F.N.Spon, Ltd., London, England, 1909.* Cloth, 8 vo., 487 pp. Price, $4.

THE CIVIL ENGINEER'S POCKET BOOK. By John C. Trautwine. 19th edition. *New York, John Wiley & Sons, 1909.* Morocco, pocket-book size, pp. xxxii + 1257 + 26. Price, $5.

Contents: Mathematics; Natural Phenomena; Mechanics, Force in Rigid Bodies; Strength of Materials; Hydrostatic Hydraulics; Constructions, etc.; Water Supply; Traction, Animal Power; Suspension Bridges; Rivets and Riveting; Railroads; Materials; Price List, etc.; Bibliography; Logarithmic Sines, etc.; Concrete.

LARGE GAS ENGINES. By Percy R. Allen. Reprinted from Cassier's Magazine, 1909. Cloth, 61 pp.

Contents: The Four-Cycle Engine—British and Continental Practice; The Four-Cycle Engine—American Practice; Two-Cycle Engines.

ENERGY. Work, Heat and Transformations. By Sidney A. Reeve, M.E. *New York, McGraw-Hill Publishing Co., 1909.* Cloth, 8 vo., 238 pp. Price, $2.

Contents: Mechanical Energy; Free and Vibratory Energies; The Mean Energetic Condition and the Energy-fund; The Two Factors of Dimensions of Energy; The Extreme or Critical Energetic Conditions; The General Nature of Mechanical Energy; What is Heat?; The Thermal Diagram; Mechanical Concepts of Thermal Phenomena, Pressure and Volume; The Two Basic Thermal Processes: Heat-transfer and Work-performance; Mechanical Concepts of Thermal Phenomena, Temperature and Entropy; The Energetic Cycle; Reversed and Irregular Cycles; Thermal Equilibrium; Transformations and Conservations.

LINSEED OIL AND OTHER SEED OILS. An Industrial Manual. By Wm. D. Ennis, M.E. *New York, D.Van Nostrand Co., 1909.* Cloth, 8 vo., xiv + 316 pp. Price, $4.

Contents: Introductory; The Handling of Seed and the Disposition of Its Impurities; Grinding; Tempering the Ground Seed and Molding the Press Cake; Pressing and Trimming the Cakes; Hydraulic Operative Equipment; The Treatment of the Oil from the Press to the Consumer; Preparation of the Cake for the Market; Oil Yield and Output; Shrinkage in Production; Cost of Production; Operation and Equipment of Typical Mills; Other Methods of Manufacturers; The Seed Crop; The Seed Trade; Chemical Characteristics of Linseed Oil; Boiled Oil; Refined and Special Oils; The Linseed Oil Market; The Feeding of Oil Cake; Miscellaneous Seed Oils; The Cotton-Seed Industry.

HENLEY'S ENCYCLOPEDIA OF PRACTICAL ENGINEERING AND ALLIED TRADES. A Practical and Indispensable Work of Reference for the Mechanical Engineer, Designer, Draftsman, Shop Superintendent, Foreman and Machinist. Edited by Joseph G. Horner, A.M.I.Mech.E. *New York, The Norman W. Henley Pub. Co., 1909.* Vol. IX, SPE-Z, 240 pp., Illustrated. Price, $6.

THE GAS ENGINE. By Cecil P. Poole. *Hill Pub. Co., New York, 1909.* Cloth, 8 vo., 6 + 97 pp., illustrated. Price, $1.

Contents: Elementary Principles; Pressures and Temperatures; Cooling and Heat Loss; Valves and Valve Gear; Ignition; Mixing Liquid Fuel with Air; Methods of Governing; Some Considerations of Design; Care and Management of Engines; Pressure, Temperature and Output Calculations.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M. E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

AMERICAN MINING CONGRESS. Report of Proceedings. Vol. 12. 1909. *Denver 1909.*

AMERICAN RAILWAY ASSOCIATION. Statistical Bulletin Nos. 60, 63, 63A. *Chicago, 1910.*

ARRANGEMENT OF ENGINE CYLINDERS TO PRODUCE UNIFORM TORQUE. Reprinted from Electrical World, November 11, 1909.

ASSOCIATION OF TRANSPORTATION AND CAR ACCOUNTING OFFICERS. *Proceedings.* December 1909. *Chattanooga, 1909.*

BIBLIOGRAPHY OF NORTH AMERICAN GEOLOGY FOR 1908, with subject index. (Bulletin 409, U. S. Geol. Survey.) By J. M. Nickles. *Washington, 1909.*

BOSTON, FINANCE COMMISSION. Report. Vol. 4. *Boston, 1909.* Gift of Samuel Whinery.

COMMERCIAL DEDUCTIONS FROM COMPARISONS OF GASOLINE AND ALCOHOL TESTS ON INTERNAL-COMBUSTION ENGINES. (Bulletin 392, U. S. Geol. Survey.) By R. M. Strong. *Washington, 1909.*

COMMISSION DE L'UNIFICATION INTERNATIONALE DES PAS DE VIS DANS LES APPAREILS D'UTILISATION DU GAZ. Second session, June, 1909.

COMPARATIVE TESTS OF RUN-OF-MINE AND BRIQUETTED COAL ON THE TORPEDO BOAT BIDDLE. (Bulletin 403, U. S. Geol. Survey.) By W. T. Ray and H. Kreisinger. *Washington, 1909.*

CONNECTICUT BUREAU OF LABOR STATISTICS. 23d report. *Hartford, 1908.* Gift of the Bureau.

CORNELL UNIVERSITY. Librarian's Report. 1908-1909. *Ithaca.*

COST KEEPING AND MANAGEMENT ENGINEERING. By H. P. Gillette and R. T. Dana. *New York-Chicago. Myron C. Clark Pub. Co., 1909.*

DEVONIAN FAUNA OF THE OURAY LIMESTONE. (Bulletin 391, U. S. Geol. Survey.) By E. M. Kindle. *Washington, 1909.*

ELECTRICAL ENGINEER'S POCKET BOOK, 1910. ed. 10. By W. H. Fowler. *Manchester, Scientific Pub. Co., 1910.*

ENGINEERS' CLUB OF CINCINNATI. Annual Address, December 16, 1909. By F. M. Crocker. *Cincinnati, 1909.*

EQUITABLE CHARGES FOR TRAMWAY SUPPLY. By H. E. Yerbury. (Institution of Electrical Engineers, 1909.) Gift of C. W. Rice.

GEOLOGICAL FEATURES OF THE COUNTRY LYING ALONG THE ROUTE OF THE PROPOSED TRANSCONTINENTAL RAILWAY IN WESTERN AUSTRALIA. (Bulletin 37, Western Australia Geol. Survey.) By C. G. Gibson. *Perth, 1909.*

GEOLOGY AND UNDERGROUND WATERS OF SOUTH DAKOTA. (Water Supply Paper No. 227, U. S. Geol. Survey.) By N. H. Darton. *Washington, 1909.*

GEOLOGY AND WATER RESOURCES OF THE NORTHERN PORTION OF THE BLACK HILLS AND ADJOINING REGIONS IN SOUTH DAKOTA AND WYOMING. (Professional Paper No. 65, U. S. Geol. Survey.) By N. H. Darton. *Washington, 1909.*

GEOLOGY OF THE LEWISTOWN COAL FIELD, MONTANA. (Bulletin 390, U. S. Geol. Survey.) By W. R. Calvert. *Washington, 1909.*

GRANITES OF VERMONT. (Bulletin 404, U. S. Geol. Survey.) By T. N. Dale. *Washington, 1909.*

HOW TO MAKE IMPROVEMENT THINNINGS IN MASSACHUSETTS WOODLANDS. By H. O. Cook. *Boston, 1910.* Gift of Massachusetts State Forester.

INCIDENTAL PROBLEMS IN GAS-PRODUCER TESTS. (Bulletin 393, U. S. Geol. Survey.) By R. H. Fernald and others. *Washington, 1909.*

LANDSLIDES IN THE SAN JUAN MOUNTAINS, COLORADO. (Professional Paper No. 67, U. S. Geol. Survey.) By E. Howe. *Washington, 1909.*

MACHINE BUILDING FOR PROFIT AND THE HARTNESS FLAT TURRET LATHE. By James Hartness. *Springfield, Vt., 1909.*

MAGNETIC SURVEY YACHT CARNEGIE AND HER WORK. From Terrestrial Magnetism, June 1909.

MANZANO GROUP OF THE RIO GRANDE VALLEY, NEW MEXICO. (Bulletin 389, U. S. Geol. Survey.) By W. T. Lee and G. H. Girty. *Washington, 1909.*

MECHANICAL ENGINEER'S POCKET BOOK. ed. 12. By W. H. Fowler. *Manchester, Scientific Pub. Co., 1910.*

MECHANICS' AND MACHINISTS' POCKET BOOK AND DIARY, 1910. By W. H. Fowler. *Manchester, Scientific Pub. Company.*

MERCURY MINERALS FROM TERLINGUA, TEXAS. (Bulletin 405, U. S. Geol. Survey.) *Washington, 1909.*

METALLOGRAPHIE. By W. Guertler. Pt. 2. *Berlin, G. Borntraeger, 1909.*

NATIONAL ELECTRIC LIGHT ASSOCIATION. (Bulletin, Vol. 3. No. 6.) *New York, 1910.* Gift of C. W. Rice.

NEW YORK CHAMBER OF COMMERCE. 51st Annual Report of the Corporation. *New York, 1909.* Gift of N. Y. State Chamber of Commerce.

NEW YORK STATE FOREST, FISH AND GAME COMMISSION. 14th and 15th Annual Reports. *Albany, 1909, 1910.* Gift of Commissioner.

NEW YORK STATE MUSEUM. 62d Annual Report. Vol. 2-4. *Albany, 1909.*

NON-MAGNETIC GAS ENGINE OF THE CARNEGIE. By J. Craig, Jr. From Terrestrial Magnetism, September, 1909.

NOTES ON SOME MINING DISTRICTS IN HUMBOLDT COUNTY, NEVADA. (Bulletin 414, U. S. Geol. Survey.) By F. L. Ransome. *Washington, 1909.*

OHIO ENGINEERING SOCIETY. Proceedings of 13th Annual Meeting. *Columbus, 1909.*

OKLAHOMA STATE UNIVERSITY. Research Bulletin. Nos. 1-2. *Norman, 1909.*

PASSENGER CAR LIGHTING. By Representatives of different Car Lighting Systems. Gift of Canadian Railroad Club.

PLEISTOCENE GEOLOGY OF THE LEADVILLE QUADRANGLE, COLORADO. (Bulletin 386, U. S. Geol. Survey.) By S. R. Capps. Jr. *Washington, 1909.*

PRIMER ON EXPLOSIVES FOR COAL MINERS. (Bulletin 423, U. S. Geol. Survey.) By C. E. Munroe and C. Hall. *Washington, 1909.*

Radioactivity of the Thermal Waters of Yellowstone National Park. (Bulletin 395, U. S. Geol. Survey.) By H. Schlundt and R. B. Moore. *Washington, 1909.*

Railway Stores Price Book. By W. O. Kempthorne. *London-New York, Spon & Chamberlain, 1909.*

Results of Spirit Leveling in West Virginia. 1896-1908, inclusive. (Bulletin 399, U. S. Geol. Survey.) By S. S. Gannett and D. H. Baldwin. *Washington, 1909.*

Slide Rule. By J. J. Clark. *Scranton, 1909.* Gift of author.

Smoley's Tables of Logarithms and Squares. ed. 5. *New York, Engineering News Pub. Co., 1908.*

Study of the Massachusetts Wood-Using Industries. By Hu. Maxwell. *Boston, 1910.* Gift of Massachusetts State Forester.

Sydney University Engineering Society, New South Wales. *Proceedings.* Vol. 13, 1908. *Sydney, 1908.*

Tests of Run-of-Mine and Briquetted Coal in a Locomotive Boiler. (Bulletin 412, U. S. Geol. Survey.) By W. T. Ray and H. Kreisinger. *Washington, 1909.*

Time and its Measurement. By James Arthur. *Chicago, 1909.* Gift of Daniel Arthur.

U. S. Geological Survey. 30th Annual Report. *Washington, 1909.*

U. S. Lighthouse Board. Annual Report. 1909. *Washington, 1909.*

Utilization of Fuel in Locomotive Practice. (Bulletin 402, U. S. Geol. Survey.) By W. F. M. Goss. *Washington, 1909.*

Valuation of Public Service Corporations. By W. H. Williams. Gift of author.

Water Resources of the Blue Grass Region, Kentucky. (Water Supply Paper No. 233, U. S. Geol. Survey.) By G. C. Matson. *Washington, 1909.*

Fourteen Miscellaneous Books. Gift of Andrew Carnegie.

EXCHANGES

American Gas Institute. *Proceedings.* Vol. 4, 1909. *1910.*

Design of Surface Condensers. By R. M. Neilson. Institution of Engineers and Shipbuilders in Scotland.

Institution of Civil Engineers. Minutes of Proceedings. Vol. 178. *London, 1909.*

Institute of Civil Engineers. Address of James Charles Inglis, President, November 2, 1909. *London, 1909.*

New Charter Suggestions submitted to the Board of Freeholders by the Board of Public Improvements, 1909. Engineers' Club of St. Louis. *St. Louis, 1910.*

Synopsis of the Report of the Superintendent of the United States Naval Observatory. 1909. *Washington, 1910.*

Worcester Polytechnic Institute. Annual Catalogue, 1909-1910. *Worcester, 1909.*

TRADE CATALOGUES

American Spiral Pipe Works, *Chicago, Ill.* Spiral riveted pipe, forged steel pipe flanges, hydraulic and exhaust steam supplies, 20 pp.

James Beggs & Co., *New York.* Feed water filtration, 32 pp.

Commercial Cable Co., *New York.* Silver anniversary souvenir, 36 pp.

Cummings Filter Co., *Philadelphia, Pa.* Water filters, 48 pp.

General Electric Co., *Schenectady, N. Y.* Bulletin 4679-A, Type DLC, commutating pole motors, 8 pp.; Bulletin 4684, Luminous arc headlight, 3 pp.; Bulletin 4706, Curve-drawing ammeter and voltmeter, 4 pp.; Bulletin 4707, Gasolene-Electric generating sets for lighting and power, 24 pp.; Bulletin 4708, Thomson direct current test meter, 4 pp.; Bulletin 4709, Portable instruments, 10 pp.; Bulletin 4713, Type F, forms K-2 and K-4 oil break switches, 7 pp.

Goldschmidt Thermit Co., *New York.* Reactions, Vol. 2, No. 4, 20 pp.

Handy Index Co., *New York.* Handy Index for January 1910, 64 pp.

Hooven, Owens, Rentschler Co., *Hamilton, O.* Bulletin 104, Type H. S. high speed Corliss engines, 10 pp.; Bulletin 105, Series A, standard girder frame Hamilton Corliss engine, 8 pp.; Bulletin 106, Hamilton-Holzwarth flexible coupling, 8 pp.; Bulletin 107, Heavy duty Hamilton Corliss engines, 8 pp.; Bulletin 108, Hamilton-Corliss compound heavy duty engines, 12 pp.; Bulletin 110, Series B Hamilton Corliss engine, 8 pp.

Jeffrey Mfg. Co., *Columbus, O.* Catalogue 69 B. Revolving, stone and gravel bell shaped, panel and tipple screens, 24 pp.; Bulletin 17, Electric mine locomotives, 68 pp.

Lamson Consolidated Store Service Co., *Boston, Mass.* Small hand and power operated elevators, dumb-waiters and automatic conveyors, 24 pp.

Murray Iron Works Co., *Burlington, Ia.* Corliss engines, 80 pp.

Niles-Bement-Pond Co., *New York.* LeBlond milling machines, 42 pp.

North Western Expanded Metal Co., *Chicago, Ill.* Expanded metal for sidewalks, culverts, slab bridges, 24 pp.

Oneida Steel Pulley Co., *Oneida, N. Y.* Catalogue of steel and wood pulleys, 48 pp.

Pierce Motor Co., *Racine, Wis.* Pierce-Racine model K, 30 h.p. motor car, 16 pp.

Pittsburgh Feed Water Heater Co., *Pittsburgh, Pa.* Feed water heater and purifier, 60 pp.

Pratt & Whitney Co., *Hartford, Conn.* Vertical surface grinder, 24 pp.

Rockwell Furnace Co., *New York.* Bulletin G, Annealing, hardening, tempering furnaces, 8 pp.

Schoen-Jackson Co., *Media, Pa.* Flexible metal tubing and connections for pressures up to 4000 pounds, 16 pp.

Stephens-Adamson Mfg. Co., *Aurora, Ill.* Conveying and Transmission, January 1910, 24 pp.

Sterling Engine Co., *Buffalo, N. Y.* High grade marine engines for cruising work and speed boats, 32 pp.

Storrs Mica Co., *Owego, N. Y.* "Never Break" mica chimneys and globes, 20 pp.

United States Mineral Wool Co., *New York.* Mineral wool in car building and steam engineering, 10 pp.

Warner Instrument Co., *Beloit, Wis.* The Auto-Meter, speed indicator for automobiles and motor cars, 24 pp.

Wheeler Condenser and Engineering Co., *Carteret, N. J.* A radical improvement in jet condensers, 15 pp.

Williamson Submarine Corporation, *Norfolk, Va.* Submarine Bulletin, December 1909, 4 pp.

UNITED ENGINEERING SOCIETY

Advertising and Selling. Vol. 19. No. 8, January 1910-date. *New York, 1910-date.*

Eagle Almanac, 1909. *Brooklyn, 1910.*

Hendricks' Commercial Register of the United States. ed. 18. *New York, 1910.*

Tribune Almanac, 1910. *New York, 1910.*

Washington Society of Engineers. By-Laws, List of Officers and Members, February 1909. Gift of Washington Society of Engineers.

World Almanac, 1910. *New York, 1910.*

GIFT OF CARNEGIE STEEL COMPANY

Data Appertaining to Light Rails and Fastenings, 1904.

Pocket Companion Containing Useful Information and Tables Appertaining to the Use of Steel. 1903.

Shapes Manufactured by Carnegie Steel Company. 1903 and supplement.

Steel Mine Timbers, Data and Tables for the use of Mining Engineers, 46 pp.

Steel Sheet Piling, 16 pp.

Schoen Steel Wheels. No. 1, 46 pp.

Schoen Steel Wheels, Designs and Specifications, 46 pp.

Carnegie Special Welding Steel, Carnegie Special Threading Steel, 40 pp.

Carnegie Steel Cross Tie and DuQuesne Rail Joint, 61 pp.

Steel Sheet Piling. Types of construction and examples of installation, 64 pp.

Steel Mine Timbers. Types of construction and examples of installation, 30 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

010 Assistant superintendent of factory manufacturing a line of small interchangeable parts in large quantities; man with technical training preferred. Must be experienced in shop management. Wanted May 1. Location, Philadelphia, Pa.

011 Wanted—Competent mechanical draftsman, preferably one who has had experience in coal mine equipment. Give full particulars, including salary required. Location, Birmingham, Ala.

012 Wanted—Thoroughly practical and energetic young man for experimental and testing department of large corporation. Must be technical graduate of three or four years standing. Principal work consists of engine and boiler testing, as well as all matters pertaining to power, exclusive of electric. Excellent chance for right man to become assistant.

013 Opening for three engineer-salesmen between twenty-five and thirty-five years of age, for selling gas and oil engines in and around New York City. Men of experience in selling are preferred. Applicant must of course be able to give the best of references.

014 Wanted—Ice-making and refrigerating machinery salesmen; experienced men preferred. Applicants should be between thirty and forty years of age.

MEN AVAILABLE

25 Technical graduate, Junior Member, several years' experience in engineering work and as sales engineer with manufacturers of internal combustion engines. Considerable traveling experience. Location of minor importance, opportunities all-important.

26 Member, at present superintendent in large machine shop, where he has been for several years, would like a change of locality; 19 years' experience in manufacture of steam engines, steam turbines, and machine tools. Capable of filling first-class position.

27 Technical graduate, experienced in varied lines of industry, has held executive positions of responsibility; desires to become associated in position of trust with good manufacturing concern, preferably located in the East or Middle West. Best of references.

28 Associate wishes position as general manager, assistant, sales manager or salesman. Good executive ability, twenty years' experience with machine-tool manufacturing company, and as appraiser and receiver.

29 Member having extensive executive and mechanical experience desires to secure a position in greater New York or vicinity as superintendent or factory manager. Light, medium or heavy lines; large experience in intricate automatic and precision mechanisms as well as modern manufacturing methods and systems.

30 Member, technical graduate; 15 years' experience in power plant equipment and rolling mill machinery with well-known firm, last five years in engineering sales work, desires responsible position in the commercial end of a metal trade business, or as branch manager on commission basis. Experience in this country and abroad.

31 Member, graduate mechanical engineer, 18 years' experience in the States, England and France, as chief draftsman, general superintendent, selling and buying engineer, desires responsible position. Good executive, specialty automatic, hydraulic and conveying machinery. Best references.

32 Graduate in mechanical engineering, Massachusetts Institute of Technology, with experience including shop, inspection and drafting work, desires position in engineer's office as assistant to superintendent, or similar position. Minimum salary expected at start, $900 per year.

33 Mechanical engineer, Associate Member; 8 years of expert work in the reduction of power costs in industrial plants. Present practice as consulting engineer in this capacity and showing large savings. Desires further connection with a limited number of manufacturing power-users, permanent position as advisory engineer with large concern operating a number of plants or commercial proposition with concern manufacturing power plant apparatus.

34 Member, with thorough business training; up-to-date factory manager; good executive and organizer; competent in manufacturing medium and light weight machinery; fully qualified to fill position of responsibility, desires a position.

35 Steam turbine designer with 5 years' experience in charge of experimental work, desires position with firm now building or intending to develop a line of steam turbines.

36 Mechanical engineer, technical graduate, 16 years' practical experience, familiar with hoisting, conveying and general mill machinery, several years experience in charge of work, desires engineering position.

37 Junior Member, graduate engineer, desires a position which will offer a future. Experienced in general manufacturing methods; at present employed as engineer and assistant to general manager of a modern plant embracing power house, pattern department, foundry, carpenter and machine shops. Salary $2000.

38 Junior, 1908, technical graduate, some knowledge of boilers, desires position testing or installing power plant apparatus.

39 Member with 15 years' experience in steam engineering and power plant equipment, drafting-room office and selling, also several years machine design; desires position with consulting engineer or in engineering work with industrial company. Location east of Pittsburg preferred.

40 Young man, ten years shop experience in one of the largest manufacturing and engineering concerns in New York, for the past five years estimating engineer in charge of contract-engineering work; technical graduate; desires position as purchasing engineer or in an engineering construction department. New England or Middle Atlantic States preferred. Can furnish the highest of references.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ALGER, Harley C. (Junior, 1908), Mech. Engr., 45 E. 16th St., Chicago Heights, Ill.

BAENDER, Fred. Geo. (Junior, 1909), Mech. Engr., 310 E. 18th Ave., Spokane, Wash.

BEECHER, J. F. (Associate, 1908), Draftsman, Pa. Steel Co., and *for mail*, 523 N. Fourth St., Harrisburg, Pa.

CALEY, Charles J. (1906), Wks. Mgr., Peterboro Lock Mfg. Co., Ltd., and Oriental Hotel, Peterboro, Ont., Canada.

CHAMBERLAIN, George E. (1907), Pres., Lewell Mfg. Co., 1416 Michigan Ave., Chicago, and *for mail*, 102 S. Waiola Ave., La Grange, Ill.

CHESTER, C. P. (Associate, 1908), Cons. Engr., 512 Caples Bldg., El Paso, Texas.

COMSTOCK, Charles Warren (Associate, 1906), Pres., Comstock-Wellman Bronze Co., 6017 Superior Ave., and *for mail*, 8803 Euclid Ave., Cleveland, O.

COOKE, Thomas F. (Junior, 1904), Cons. Engr., Webb & Cooke, 338 Ellicott Sq., and *for mail*, 618 Delaware Ave., Buffalo, N. Y.

FULWEILER, John Edwin (Junior, 1908), United Gas Improvement Co., and *for mail*, 4335 Chestnut St., Philadelphia, Pa.

GUMP, Walter B. (Junior, 1902), Mech. Engr., 2510 Juliet St., Los Angeles, Cal.

INGALLS, Fred. D. B. (1909), Cons. Mech. Engr., Rosenblum Bldg., 106 E. Fayette St., Syracuse, N. Y.

JOHNSON, Paul F. (1905), Johnson Service Co., Milwaukee, Wis.

KEITH, Thomas M. (Junior, 1905), Robins Conveying Belt Co., 30 Church St., New York, N. Y.

LAIDLAW, Walter (1889), Manager, 1905–1908; Intl. Steam Pump Co., 115 Broadway, New York, N. Y.

LEWIS, Wm. Yorath (1902), Engr., 49 Queen Victoria St., London, E. C., England.

MacKENZIE, Donald (Junior, 1902), Swift & Co., Stock Yards Sta., Chicago, Ill.

MORRISON, Clarke J. (1909), 52 E. 19th St., New York, N. Y.

NEILER, Samuel Graham (1907), Cons. Engr. and Pres., Pierce, Richardson & Neiler, 1407–1411 Manhattan Bldg., Chicago, and Oak Park, Ill.

NIBECKER, Karl (Junior, 1908), Mech. Engr., Southwark Fdy. & Mch. Co., Fifth and Washington Ave., Philadelphia, and *for mail*, Glen Mills, Pa.

POTTS, S. Warren (1909), 628 W. 148th St., New York, N. Y.

SCHAKEL, Jacob Daniel (Associate, 1907), Otis Elev. Co., Northland Ave. and Girder St., Buffalo, N. Y.

SCHREUDER, Andrew M. (1898; 1909), Supt., Philadelphia Textile Mchy., Co., Hancock and Somerset Sts., Philadelphia, and *for mail*, 5351 Wayne Ave., Germantown, Philadelphia. Pa.

SCOTT, Walter G. (Junior, 1909), Allis-Chalmers Co., and *for mail*, University Club, West Allis, Wis.

SPENCER, Frank C. (Associate, 1908), Mech. and Constr. Engr., 5258 Indiana St., Chicago, Ill.

STARBUCK, George F. (Junior, 1901), Draftsman, Boston Elev. Ry., Boston, and *for mail*, Waltham, Mass.

SMITH, S. H. (Associate, 1907), Supt., North Melbourne Elec. Tramways & Ltg. Co., Ltd., Mt. Alexander Rd., and Clydehall, Harding and East Sts., Ascot Vale, Melbourne, Australia.

TAYLOR, Cecil Hamelin (Associate, 1908), Ch. Engr., Hudson Motor Car Co., and Pasadena, Detroit, Mich.

TREGELLES, Henry (1888), Bartolome Mitre 544, Buenos Aires, and Hurlingham and Pacifico, Buenos Aires, Argentine Repub., South America.

VON AMMON, Siegfried (1904; 1905), Fontella, Va.

WADDELL, Charles E. (1903; 1907), Cons. Engr., 78 Patton Ave., Asheville, N. C.

WALKER, Gilbert S. (1904), Isthmian Canal Com., Mills Bldg. Annex, Washington, D. C.

WHEELER, Earl (Junior, 1907), Elec. and Mech. Engr., Elec. Speedometer & Dynamometer Mfg. Co., 1317 New York Ave., and *for mail*, The Benedict, 1810 I St., N. W., Washington, D. C.

WHEELER, Wm. Trimble (1905), 286 Greenwich St., and 340 W. 21st. St., New York, N. Y.

NEW MEMBERS

ARBOGAST, Victor R. (1909), Wks. Engr. and Supt., Natl. Radiator Gesellschaft, Schoenebeck Elbe, Germany.

BACON, John Lord (1899; 1909), Engr. and Supt. of Constr., R. P. Shields & Son, 605 Scripps Bldg., and *for mail*, 3576 A St., San Diego, Cal.

BLUM, Arthur N. (1909), Asst. Mgr., Sormovo Engrg. Wks., Nijni Novgorod, Russia.

BORNHOLT, Oscar Charles (1904; 1909), Mech. Engr., Ford Motor Co., Piquette Ave. and Beaubien St., and *for mail*, 50 Philadelphia Ave., Detroit, Mich.

DAY, Leonard A. (1909), First Asst. Mech. Engr., St. Louis Water Dept., and *for mail*, 4015 Greer Ave., St. Louis, Mo.

DIMAN, W. G. (1909), Senior Engr. Officer, U. S. S. Mayflower, Navy Dept., Washington, D. C.

FERRIER, Joseph J. (Junior, 1909), So. Pacific Co., 1110 Flood Bldg., San Francisco, Cal.

FRANK, Edwin (Junior, 1909), Designer, Maffei-Schwartzkopff Wks., G. m. b. H., and *for mail*, Kaiser Wilhelmstr., 10, Zeuthen i M, Germany.

HERRIMAN, Victor D. (Junior, 1909), Engr., Intl. Steam Pump Co., 115 Broadway, New York, and *for mail*, 167 Quincy St., Brooklyn, N. Y.

HORTON, Charles M. (Junior, 1909), Secy., Ford Refrig. Air-Machine Co., and *for mail*, 101 W. 101st St., New York, N. Y.

KNEELAND, Frank H. (Junior, 1909), Mech. Engr., U. S. Coal & Coke Co., Gary, W. Va.

PARSONS, Edmund S. (Junior, 1909), Mech. Engr., Remington Typewriter Wks., and *for mail*, 56 West St., Ilion, N. Y.

POLHEMUS, Louis Edward (Junior, 1909), Asst. M.M., Mexican Light & Power Co., Necaxa (Estado de Puebla), Mexico.

POOLE, Cecil P. (1909), Joint Editor, Power and The Engineer, 505 Pearl St., New York, N. Y., and South Orange, N. J.

RANSOM, T. Wells (1909), Cons. Mech. Engr., Board of Public Wks., San Francisco, Cal.

SOVERHILL, Harvey A. (1909), Supt., Root & Van Dervoort Engrg. Co., East Moline, and 623 23d St., Moline, Ill.

TORRANCE, Chas. Everett (Junior, 1909), Instr., Sibley College, and *for mail*, 638 Stewart Ave., Ithaca, N. Y.

DEATHS

BALDWIN, Stephen W.
BATCHELOR, Charles.
SANGUINETTI, Percy A.

GAS POWER SECTION

CHANGES OF ADDRESS

BAENDER, Fred Geo. (1909), Mem. Am.Soc.M.E.
FISKE, Geo. Wallace (Affiliate, 1909), 610 W. 10th St., Topeka, Kan.
MYERS, Theodore B. (Affiliate, 1909), Woodcliff-on-Hudson, N. J.
PARKER, Lewis C. (Affiliate, 1908), present address unknown.

NEW MEMBERS

COLLINS, Harold W. T. (Affiliate, 1910), Designer, Lodge & Shipley Mch. Tool Co., Cincinnati, and *for mail*, 2242 Cameron Ave., Norwood, O.
CRAIG, James (1910), Mem. Am.Soc.M.E.
DORSEY, Howard Alex. (Affiliate, 1910), Instr. Mech. Engrg., Univ. of Cincinnati, Cincinnati, O.
FERRIER, Joseph J. (1910), Mem. Am.Soc.M.E.
GARDNER, F. M. (Affiliate, 1910), Engr. and Salesman, Fairbanks, Morse & Co., and *for mail*, 137 W. Fourth St., Cincinnati, O.
GRIFFITHS, Leonard L. (1910), Mem. Am.Soc.M.E.
JEWETT, Arthur C. (1910), Mem. Am.Soc.M.E.
MANGELSDORFF, Max F. (Affiliate, 1910), 115 Nassau St., New York, N. Y.
POOLE, Cecil P. (1909), Mem.Am.Soc.M.E.
READ, Carleton A. (1910), Mem. Am.Soc.M.E.
ROE, Joseph W. (1910), Mem. Am.Soc.M.E.
SCHWENKER, Robert Frederick (Affiliate, 1910), Mech. Engr., 3913 Regent Ave., Norwood, O.
TORRANCE, Charles E. (1910), Mem. Am.Soc.M.E.
WHITTLESEY, James Thomas (1910), Mem. Am. Soc. M. E.

STUDENT BRANCHES

CHANGES OF ADDRESS

DUNSHEATH, L. M. (Student, 1909), 105 E. Healey St., Champaign, Ill.
GROSSBERG, Arthur S. (Student, 1909), Mineral Point Zinc Co., Depue, Ill.
HERBERT, E. H. (Student, 1909), Doak Gas Eng. Co., 7-9 First St., San Francisco, Cal.
KOWALEWSKI, A. J. (Student, 1910), 582 Main Bldg., State College, Pa.
MANSFIELD, W. M. (Student, 1909), 2924 Mt. Vernon Ave., Milwaukee, Wis.
ROMIG, F. G. (Student, 1910), 601 S. Wright St., Champaign, Ill.
TIFFT, R. H. (Student, 1909), 65 Park Avenue, New York, N. Y.

NEW MEMBERS

COLUMBIA UNIVERSITY

BAUM, A. L. (Student, 1910), 252 W. 128th St., New York, N. Y.
BLUMENFELD, Ralph (Student, 1910), 508 W. 114th St., New York, N. Y.
BRETTELL, C. (Student, 1910), 29 Meadow Lane, New Rochelle, N. Y.
FRAMBACH, F. S. (Student, 1910), 430 W. 119th St., New York, N. Y.
GATELY, W. A. (Student, 1910), 125 E. 54th St., New York, N. Y.
GUITERAS, J. G. (Student, 1910), 1 Livingston Ave., Yonkers, N. Y.
HAYNES, J. L. (Student, 1910), 3216 Glenwood Rd., Brooklyn, N. Y.
JAROS, A. L. (Student, 1910), 542 W. 112th St., New York, N. Y.
KATZ, E. J. (Student, 1910), 249 E. 68th St., New York, N. Y.
KIRSCHBERG, M. (Student, 1910), 25 W. 123d St., New York, N. Y.
LACY, F. T. (Student, 1910), 411 W. 115th St., New York, N. Y.
LORD, J. W. (Student, 1910), 163 E. 71st St., New York, N. Y.

CORNELL UNIVERSITY

BENBOW, J. R. (Student, 1910), 210 Linden Ave., Ithaca, N. Y.
HAM, C. W. (Student, 1910), 126 E. Seneca St., Ithaca, N. Y.
PIMPER, T. F. (Student, 1910), 427 E. Seneca St., Ithaca, N. Y.
ROBINSON, G. E. (Student, 1910), 208 Williams St., Ithaca, N. Y.
STURGIS, R. F. (Student, 1910), 110 Sage Pl., Ithaca, N. Y.

BROOKLYN POLYTECHNIC INSTITUTE

BARRETT, S. A. K. (Student, 1910), 114 Pierrepont St., Brooklyn, N. Y.
ERICSON, E. O. (Student, 1910), Helmetta, Middlesex Co., N. J.
GRIFFIN, E. F. (Student, 1910), Box 417, Oyster Bay, L. I., N. Y.

PENNSYLVANIA STATE COLLEGE

FORKER, Geo. M. (Student, 1910), 203 McAllister Hall, State College, Pa.
KAIER, John B. (Student, 1910), 283 Lehigh St., Wilkes-Barre, Pa.
MARSH, Karl H. (Student, 1910), The Lincoln, Youngstown, O.
PERHAM, Dean E. (Student, 1910), 512 Main Bldg., State College, Pa.
WESTERMAN, John H. (Student, 1910), Theta Psi House, State College, Pa.

STATE AGRICULTURAL COLLEGE OF OREGON

GRAF, Samuel Herman (Student, 1910), State Agri. College of Oregon, Corvallis, Oregon.
HASKELL, William Dexter (Student, 1910), State Agri. College of Oregon, Corvallis, Oregon.
LINES, J. Donald (Student, 1910), State Agri. College of Oregon, Corvallis, Oregon.

STEVENS INSTITUTE OF TECHNOLOGY

BRUCE, A. C. (Student, 1910), 934 Bloomfield St., Hoboken, N. J.
MONESTEL, Alberto A. (Student, 1910), 518 Hudson St., Hoboken, N. J.
SCHOCH, Floyd W. (Student, 1910), 507 River St., Hoboken, N. J.

UNIVERSITY OF ILLINOIS

BANNISTER, B. (Student, 1910), 412 E. Green St., Champaign, Ill.
HASBERG, Will (Student, 1910), 307 E. Green St., Champaign, Ill.
MURDUCK (Student, 1910), 705 W. Hills St., Champaign, Ill.

UNIVERSITY OF KANSAS

FAIRCHILD, F. P. (Student, 1910), 946 Ohio St., Lawrence, Kan.

COMING MEETINGS

MARCH–APRIL

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN ASSOCIATION OF RAILROAD SUPERINTENDENTS
March 18, Chicago. Secy., O. G. Fetter.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
March 11, with Am. Soc. M. E., 29 W. 39th St., New York. Papers: Electric Mine Hoists with Illgner Motor Generator Set, R. R. Seeber; Comparison of Electric and Compressed Air Drives for Mine Hoists, W. Sykes. March 30-April 1, Selwyn Hotel, Charlotte, N. C. Papers: Economics of Hydroelectric Plants, W. S. Lee, Mem. Am. Soc. M. E., Electric Drive in Textile Mills, A. Milmow; Gas Engines in City Railway and Light Service, E. D. Latta, Jr.; Protecting Insulators from Lightning and Power Arc Effects On Lines of the Niagara and Lockport Power Co., L. C. Nicholson. Secy., R. W. Pope. April 21, San Francisco, Cal. Papers : Economics of a Generator Power System, P. M. Downing; Hydroelectric Developments and Irrigation, J. C. Hays.

AMERICAN INSTITUTE OF MINING ENGINEERS
March 1-5, Spring Meeting, Hotel Schenley, Pittsburg, Pa. Secy., R. W. Raymond, 29 W. 39th St., New York.

AMERICAN MATHEMATICAL SOCIETY
April 30, Columbia University, 150 W. 116th St., New York. Secy., F. N. Cole.

AMERICAN RAILWAY ENGINEERING ASSOCIATION
March 14-17, Chicago. Secy., E. H. Field, Monadnock Bldg.

AMERICAN RAILWAY ENGINEERING AND MAINTENANCE OF WAY ASSOCIATION
March 15–17, annual convention, Chicago. Secy., E. H. Fritch, 962 Monadnock Blk.

AMERICAN SOCIETY OF CIVIL ENGINEERS
March 2, 1910, 220 W. 57th St., New York. Paper: The Improved Water and Sewage Works of Columbus, O., J. H. Gregory. Secy., C. W. Hunt.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
March 8, 29 W. 39th St., New York. March 11, Auditorium Edison Electric Illuminating Co. of Boston, Boston, Mass. May 31-June 3, Spring Meeting, Atlantic City, N. J. July 26-29, meeting in Birmingham, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SUPPLY AND MACHINERY MANUFACTURERS ASSOCIATION

SOUTHERN SUPPLY AND MACHINERY DEALERS ASSOCIATION

April 5–7, Seminole Hotel, Jacksonville, Fla.

AMERICAN WATERWORKS ASSOCIATION

April 26-30, annual convention, New Orleans, La. Secy., J. M. Diven, 14 George St., Charlestown, S. C.

BROOKLYN ENGINEERS CLUB

March 3, monthly meeting, 117 Remsen St. Paper: The Making of a Marble Quarry, by T. B. Hamilton. Secy., Joseph Strachan.

BOSTON SOCIETY OF ARCHITECTS

March 1, regular meeting with reports, Parker House. Secy., E. J. Lewis, Jr.

BOSTON SOCIETY OF CIVIL ENGINEERS

March 2, annual meeting, Boston City Club, 7.30 p.m. Paper: Ventilation of Subways, G. A. Soper.

CANADIAN FORESTRY ASSOCIATION

March 10-11, Fredericton, N. B. Secy., Jas. Lawler, 11 Queen's Park, Toronto, Ont.

CANADIAN FREIGHT ASSOCIATION

April 14, annual meeting, Montreal. Secy., T. Marshall, Toronto, Ont.

CANADIAN MINING INSTITUTE

March 2-4, annual meeting, Toronto, Ont. Secy., H. Mortimer-Lamb, Windsor Hotel, Montreal.

ENGINEERS SOCIETY OF WESTERN PENNSYLVANIA

March 1, Fulton Bldg., Pittsburg, 8 p.m. Discussion on Present-Day Needs in Structural Materials. Secy., E. K. Hiles.

FLORIDA ELECRTIC LIGHT AND POWER ASSOCIATION

April 12, annual meeting, Tampa. Secy., G. I. Doig, Gainesville.

IOWA ASSOCIATION CEMENT USERS

March 9-11, Cedar Rapids. Secy., Ira Williams, Ames.

IOWA ELECTRICAL ASSOCIATION

April 20-21, annual convention. Secy., W. N. Keiser, Des Moines.

IOWA STREET AND INTERURBAN RAILWAY ASSOCIATION

April, Sioux City. Secy., L. D. Mathes.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY, Student Branch, Am. Soc. M. E.

March 8, annual meeting, Boston. Secy., A. P. Truette.

MINNESOTA ELECTRIC ASSOCIATION

March, St. Paul. Secy., B. W. Cowperthwait.

MISSOURI ELECTRIC, GAS, RAILWAY AND WATERWORKS ASSOCIATION.

April 14-16, Jefferson City. Secy., C. L. Clary, Sikeston.

MODERN SCIENCE CLUB

March 29, annual dinner; April 12, annual election, 125 S. Elliott Pl., Brooklyn, N. Y. Secy., J. A. Donnelly.

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS

April 27–28, semi-annual meeting. Secy., Dr. C. J. H. Woodbury, Mem. Am. Soc. M. E., Box 3672, Boston.

NATIONAL MACHINE TOOL-BUILDERS ASSOCIATION
May, Spring Convention, Rochester, N. Y. Secy., C. Hildreth, Worcester, Mass.

NEW ENGLAND RAILROAD CLUB
March 8, annual meeting, Copley Square Hotel, Boston. Subject for Discussion: M. C. B. Rules of Interchange. Secy,. G. H. Frazier, 10 Oliver St.

NEW ENGLAND STREET RAILWAY CLUB
March 24, annual meeting, Boston, Mass. Secy., J. J. Lane, 12 Pearl St.

NEW ENGLAND WATERWORKS ASSOCIATION
April 13, special meeting, Hartford, Conn. June, Providence, R. I. September 14-16, annual convention, Rochester, N. Y. Secy., Willard Kent, Narrangansett Pier, R. I.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
March 22, April 26; West Hall, R. I. School of Design, 8 p. m. Papers: The Fitchburg Plan of Coöperative Industrial Education, W. B. Hunter, M. A. Coolidge; Oxy-Acetylene Welding and Cutting, Henry Cave. Secy., Prof. T. M. Phetteplace, Mem. Am. Soc. M. E., 48 Snow St.

RAILWAY SIGNAL ASSOCIATION
March 14, Chicago. Secy., C. C. Rosenberg, Bethlehem, Pa.

SOCIETY OF CHEMICAL INDUSTRY
April 1, annual meeting, New England section. Secy., Alan Claflin, 88 Broad St., Boston, Mass.

STEVENS ENGINEERING SOCIETY
March 1, 8, 15, 22, 31, Hoboken, N. J. Papers: Automobiles, J. F. O'Rourke, H. F. Cuntz; Arts and Industries of the Orient, W. J. Hammer, C. R. Richards; on the Gyrostat and its Applications, G. V. Wendell. Secy., R. H. Upson.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
March			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engineers and Architects	Z. deNémeth	8.30
8	The American Society of Mechanical Engineers	Calvin W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute of Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of the City of New York	C. D. Pollock	8.15
April			
2	Amer. Soc. Hungarian Engineers and Architects	Z. deNemet	8.30
6	Wireless Institute	S. L. Williams	7.30
7	Blue Room Engineering Society	W. D. Sprague	8.00
8	American Institute of Electrical Engineers	R. W. Pope	8.00
12	The American Society of Mechanical Engineers	Calvin W. Rice	8.15
14	Illuminating Engineering Society	P. S. Millar	8.00

Date	Society	Secretary	Time
April			
15	New York Railroad Club	H. D. Vought	8.15
19	New York Telephone Society	T. H. Lawrence	8.00
27	Municipal Engineers of the City of New York	C. D. Pollock	8.15

OFFICERS AND COUNCIL

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Geo. M. Bond .... Hartford, Conn.
R. C. Carperter .... Ithaca, N. Y.
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Terms expire at Annual Meeting of 1910

Charles Whiting Baker .... New York
W. F. M. Goss .... Urbana, Ill.
E. D. Meier .... New York

Terms expire at Annual Meeting of 1911

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Members of the Council for 1910

John R. Freeman .... Providence, R. I.
Frederick W. Taylor .... Philadelphia, Pa.
F. R. Hutton .... New York
M. L. Holman .... St. Louis, Mo.
Jesse M. Smith .... New York

MANAGERS

Wm. L. Abbott .... Chicago, Ill.
Alex. C. Humphreys .... New York
Henry G. Stott .... New York

Terms expire at Annual Meeting of 1910

H. L. Gantt .... Pawtucket, R. I.
I. E. Moultrop .... Boston, Mass.
W. J. Sando .... Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

J. Sellers Bancroft .... Philadelphia, Pa.
James Hartness .... Springfield, Vt.
H. G. Reist .... Schenectady, N. Y.

Terms expire at Annual Meeting of 1912

TREASURER

William H. Wiley .... New York

CHAIRMAN OF THE FINANCE COMMITTEE

Arthur M. Waitt .... New York

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F. R. Hutton .... New York

SECRETARY

Calvin W. Rice .... 29 West 39th Street, New York

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F. R. Hutton
H. L. Gantt
F. M. Whyte

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Waldo H. Marshall (4)

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Bernard V. Swenson (2)
Edward Van Winkle (4)
H. R. Cobleigh (5)

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Ambrose Swasey (1)
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MEETINGS

Willis E. Hall (5), *Chairman*
L. R. Pomeroy (2)
Wm. H. Bryan (1)
Chas. E. Lucke (3)
H. de B. Parsons (4)

MEMBERSHIP

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PUBLICATION

D. S. Jacobus (1) *Chairman*
H. W. Spangler (3)
H. F. J. Porter (2)
Geo. I. Rockwood (4)
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RESEARCH

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R. C. Carpenter (1)
Ralph D. Mershon (3)
Jas. Christie (5)

Note—Numbers in parentheses indicate number of years the member is yet to serve.

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
A. P. Trautwein — Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet — H. H. Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman* — F. R. Hutton
G. M. Basford — D. S. Jacobus
Jesse M. Smith

On Conservation of Natural Resources

Geo. F. Swain, *Chairman* — L. D. Burlingame
Charles Whiting Baker — M. L. Holman
Calvin W. Rice

On International Standard for Pipe Threads

E. M. Herr, *Chairman* — Geo. M. Bond
William J. Baldwin — Stanley G. Flagg, Jr.

On Standards for Involute Gears

Wilfred Lewis, *Chairman* — E. R. Fellows
Hugo Bilgram — C. R. Gabriel
Gaetano Lanza

On Power Tests

D. S. Jacobus, *Chairman* — L. P. Breckenridge — Edward F. Miller
Edward T. Adams — William Kent — Arthur West
George H. Barrus — Charles E. Lucke — Albert C. Wood

On Student Branches

F. R. Hutton, Honorary Secretary

On Meetings of the Society in Boston

Ira N. Hollis, *Chairman* — I. E. Moultrop, *Secretary*
Edward F. Miller — J. H. Libbey
Charles T. Main

On Meetings of the Society in St. Louis

Wm. H. Bryan, *Chairman* — Ernest L. Ohle, *Secretary*
M. L. Holman

SOCIETY REPRESENTATIVES

1910

On John Fritz Medal

Ambrose Swasey (1) — Chas. Wallace Hunt (3)
F. R. Hutton (2) — Henry R. Towne (4)

On Board of Trustees United Engineering Societies Building

F. R. Hutton (1) — Fred J. Miller (2)
Jesse M. Smith (3)

On Library Conference Committee

J. W. Lieb, Jr., Chairman of the Library Committee, Am. Soc. M. E.

On National Fire Protection Association

John R. Freeman — Ira H. Woolson

On Joint Committee on Engineering Education

Alex. C. Humphreys — F. W. Taylor

On Government Advisory Board on Fuels and Structural Materials

Geo. H. Barrus — P. W. Gates
W. F. M. Goss

On Advisory Board National Conservation Commission

Geo. F. Swain — John R. Freeman
Chas. T. Main

On Council of American Association for the Advancement of Science

Alex. C. Humphreys — Fred J. Miller

Note—Numbers in parentheses indicate number of years the member is yet to serve.

OFFICERS OF THE GAS POWER SECTION

1909

CHAIRMAN
J. R. Bibbins

SECRETARY
Geo. A. Orrok

GAS POWER EXECUTIVE COMMITTEE

F. H. Stillman, *Chairman*
F. R. Hutton
G. I. Rockwood
H. H. Suplee
F. R. Low

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H. V. O. Coes
A. E. Johnson
F. S. King
A. F. Stillman
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George W. Whyte
S. S. Wyer

GAS POWER MEETINGS COMMITTEE

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C. W. Obert
W. H. Blauvelt
C. T. Wilkinson

GAS POWER LITERATURE COMMITTEE

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G. D. Conlee
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L. V. Goebbels
L. N. Ludy
L. S. Marks
T. M. Phetteplace
G. J. Rathbun
S. A. Reeve
A. L. Rice
A. J. Wood

GAS POWER INSTALLATIONS COMMITTEE

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J. R. Bibbins
E. T. Adams
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H. F. Smith
Louis (. Doelling

OFFICERS OF STUDENT BRANCHES

STUDENT BRANCH	AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	....	C. F. Hirshfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	G. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	E. A. Rogers	H. C. Warren
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. W. Templin	H. A. Houston
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	C. E. Johnson	C. A. Swiggett
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	W. F. Colman	S. G. Wood
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THE TESTING OF WATER WHEELS AFTER INSTALLATION

By Prof. C. M. Allen, Worcester, Mass.

Member of the Society

In the last few years there has been a growing demand for brake tests of water wheels after installation, the object being to determine the horsepower and in many cases the efficiency of the wheels, under actual running conditions, as well as to ascertain whether the wheels are up to their guaranteed rating.

2 The Holyoke testing flume is the only place in the United States where commercial tests of water wheels are made. For purposes of comparison under similar conditions, these tests serve their purpose well and their influence has been great in the development of the modern efficient turbine; but however well a wheel may show up under test made as just described, it may or may not give equally good results after installation. That depends entirely upon the kind of wheel, conditions of setting, and requirements of performance.

3 If the wheel is given a good setting and is allowed to run at the proper speed under a head suited to the design, then it will perform its rated work, which can be accurately computed from the original tests made at Holyoke, provided the wheel is not too large for the testing flume. This flume was not designed to test the largest of our modern turbines, nor is it suitable for testing high-head turbines. If the wheel is not given a fair setting and is required to run at too high a speed (which seems to be almost universal practice), it will fall down on both power and efficiency, the drop in each depending upon the departure from the normal conditions.

4 The efficiency of water wheels under actual working conditions has a very direct bearing upon the conservation of natural resources, and every inducement should be offered to keep that efficiency high. The water wheel as it is leads all other prime movers in efficiency.

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Under ideal working conditions of the steam engine, gas engine and steam turbine, the water wheel has at least three times as good an efficiency as the best of these. If, therefore, by increasing the efficiency of the water wheel, even by a small percentage, we are able to get just so much more power from the same amount of water used, this clearly has a direct bearing on the question of conservation.

5 There is one difference between coal and water, considered as sources of power, which is of more importance than is usually given it, namely: that if water is not used for power, and used efficiently, then that power is lost forever. It is a case of use or lose. The coal not mined or used still remains for the years to come, but the water power not used at all, or not used efficiently, is gone. "The mill will never grind again with the water that is past." As a matter of fact, there are many plants today operated by water turbines that are from 10 to 15 per cent lower in efficiency than they might have been, had the proper kind of installation and setting been definitely known and used. It is, therefore, the desirability of determining the proper setting of the turbine, and the best speed of operation under actual running conditions, that has created, in a large measure, the demand for brake testing after installation.

6 The importance of having a water power developed and operated with maximum efficiency needs no argument; yet of the several reasons that may be mentioned, one of the most important, though apparently not always considered, is purely financial. In the case of several typical hydro-electric installations, for instance, the cost of power house, dam, reservoirs generators, transmission lines, etc., is over 90 per cent of the total, leaving less than 10 per cent for the wheels; but upon the performance of the wheels, depends to a large extent the income from the entire installation. Any increase in the efficiency of these wheels means a direct gain in the power output, and this means, or should mean, not only bigger dividends but also a probable saving of coal somewhere.

7 Wheels have been tested in the last few years in several plants where the difference between the guaranteed efficiency which should have been obtained, and the actual efficiency due to poor settings, was enough to make the difference between a good paying investment and one that did not pay at all.

8 When wheels are installed in hydro-electric stations, and apparently do not show the power and efficiency guaranteed, the question naturally arises, why should a brake test be thought necessary? This may be answered in several ways. In the author's opinion, an

electrical test properly conducted should be sufficiently accurate and reliable in determining the output of the wheels. The reason for making a good many brake tests in the past has been to settle disputes between the hydraulic power and the electrical interests, relative to the guaranteed operation of the plant. The majority of water wheel builders in this country are not willing to abide by the results of an electrical test unless the wheels show up to the guaranteed power by such tests. The generator manufacturers are also unwilling to assume that the generators are low in efficiency, or that their testing apparatus is unreliable. The use of the brake for actually determin-

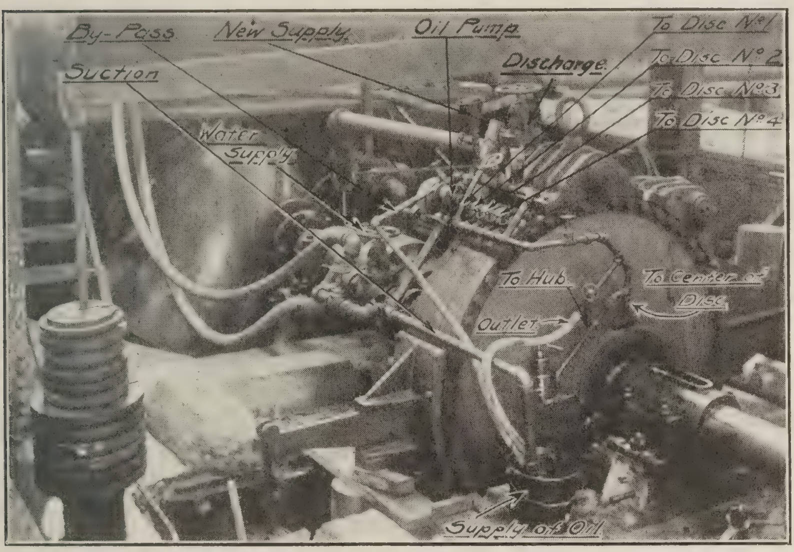

FIG. 1 60-IN. FOUR-DISC DYNAMOMETER; CAPACITY 3000 H.P. AT 200 R.P.M.

ing the horse-power-output of the wheels at the generator coupling is satisfactory to all parties concerned, for the simple reason that the apparatus is very much less complicated and more easily understood. Moreover, the accuracy of the brake can be determined on the ground while under test, as the calibration of the machine can be made at that time, and there is no possible chance for a serious error. In other words, the brake test is the simplest, most accurate, and most direct method of measuring power, and is universally recognized as the standard.

9 In making electrical tests there are many more chances for errors to creep in than in making the brake tests. Ordinarily

several electrical instruments are needed. which should be carefully calibrated before and after tests. These are liable to become changed in transit to the station. Many times they are used under different conditions of temperature, of magnetism, of connection, etc., than when calibrated, and the total results are liable to error on account of the number of instruments to be read, thus bringing in errors which may be more or less cumulative.

10 There is another reason for making brake tests rather than electrical tests at the present time, which is purely a human one. It is that no one but an electrical engineer, or some one with consider-

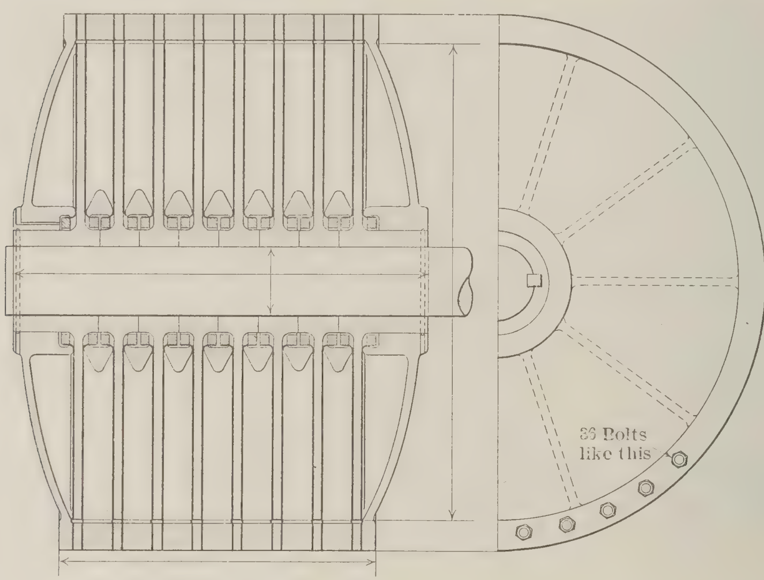

Fig. 2 Assembly Drawing of 28-in. Alden Absorption Dynamometer; Capacity 2000 h.p., 1800 r.p.m.; Eight Discs

able electrical engineering training, can understand the method used on a complete electrical test, while everyone interested in the plant can understand the method of the mechanical brake test. All parties interested can have their representatives on the ground, to check up all the measurements on the brake and calibrate the dynamometer exactly as it is used under running conditions, and so get with certainty the output of the wheels which is to be delivered to the generator. Furthermore, in order to determine the complete characteristics of the wheels under varying gate openings, and with any considerable variation in speed, it is not always practical to use an electrical generator to furnish the load.

11 It was with the idea of meeting this demand that the Alden absorption dynamometer has been developed and built in large sizes. The principle of the dynamometer is so familiar that only a brief description will be given. It is a form of Prony brake, and usually consists of several smooth circular revolvable cast-iron discs (See Fig. 2), keyed to the shaft which transmits the power; a non-revolvable housing having its bearings upon the hubs of the revolving discs; and a pair of thin copper plates in contact with each cast-iron disc, the plates being integral with the housing. Through a system of piping, water under pressure is circulated through chambers between the units, each consisting of a disc and its copper plates, and between

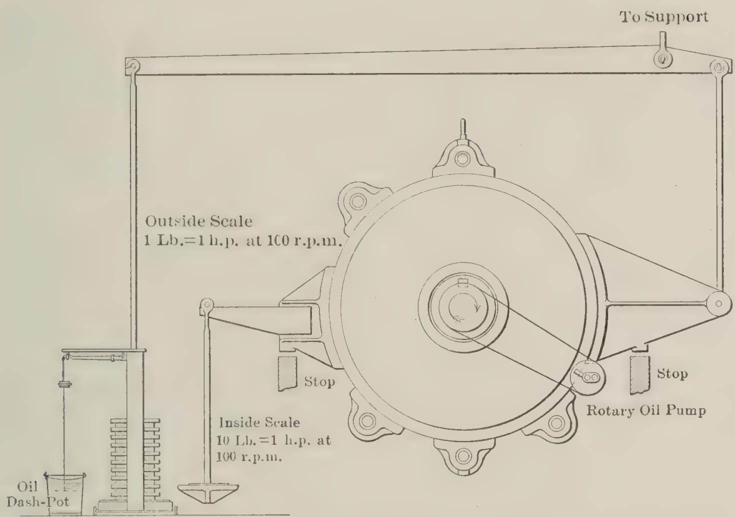

FIG. 3 SKETCH OF DYNAMOMETER SHOWING WORKING PRINCIPLES

the outer plate at either end and the wall of the housing. The water pressure is regulated by hand or by an automatic valve. Another system of piping circulates oil for lubricating the surface of the copper plates next to the revolving discs. In the large-sized machines, oil is impelled by a belt-driven pump mounted on the housing, enters the chambers at the circumference and is forced along the radial grooves of the discs to the hub, and completes its circuit through hose connections to the pump.

12 The power required to drive the pump is measured with, and included in, the power of the dynamometer, for the pump is bolted to the housing, and the driving tension in the belt which operates

the pump tends to rotate the housing in the same manner as does the internal friction of the discs; this makes a calibration to determine power used by the oil pump unnecessary (See Fig. 3).

13 When the dynamometer is in use, water passes through the chambers of the housing and between the several units of plates and discs, and by its pressure tends to force the plates against the sides of the revolving discs. This pressure increases the friction between the discs and plates, and this friction offers resistance to the rotation of the discs. The construction resembles that of a constantly slipping friction disc clutch. The resistance to turning imposed by the friction plates and discs is balanced by the weighing apparatus.

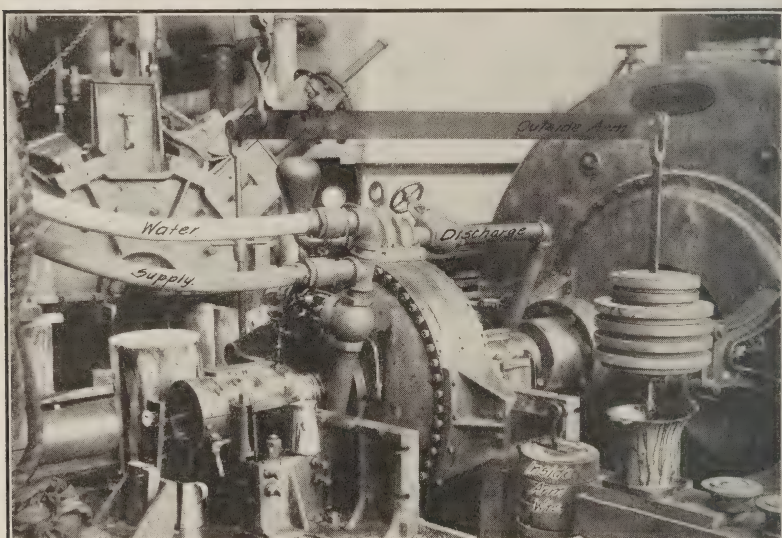

Fig. 4 Method of Counterbalancing Under Load

14 The power transmitted from the wheel under test tends to rotate the housing. This tendency is counteracted by the dead weights or a platform scales, and the housing is kept from rotating, beyond prescribed limits, by stops on either side of a lever arm bolted to the housing. The weighing apparatus by which the power absorbed is measured is delicately adjusted on knife-edge bearings. There are two sets of lever scales, which may be called the outside and inside scales. The outside indicates 1 h.p. for 1 lb. weight per 100 r.p.m. The inside scale indicates 1 h.p. for 10 lb. weight per 100 r.p.m. The outside scale serves not only to assist in balancing

the load—that is, to weigh it—but also to take the weight of the housings from the bearings on the hub of the revolving discs. (See Fig. 3 and Fig. 5.)

15 It is possible to take from the bearings not only the weight of the dynamometer, but also the weight of the shaft. There have recently been made several tests on wheels developing over 2000 h.p., where the entire weight of the dynamometer and shaft (about 13,000 lb. total) was counterbalanced so nicely that the nearest required running bearing was that of the water wheel some seven feet away from the dynamometer. In other words, when the load was on the

Fig. 5 Auxiliary Method of Counterweighting Under Load

dynamometer, it was as if it were placed on an overhanging shaft about seven feet from the nearest bearing. This point is an interesting one in mechanics, and shows that the wheels tested in place can be given a fair treatment under test, in that no additional load due to the weight of the dynamometer is put upon the wheel bearings.

16 To calibrate the dynamometer requires simply the determination of the distance from the center of the shaft to the knife edge bearing of the lever rod, and the ratio of the overhead lever; and the standardization of the dead weight, if used directly, or of the platform scales. Besides this, it is necessary to determine the initial load on

the dynamometer due to the unbalanced effect of the piping, fittings, arms, stops, lever and scale pan. This should be done at the time of test and with the apparatus as used. The usual method employed (shown in Fig. 7) consists in disconnecting the shaft coupling and raising the dynamometer so that parallel irons can be placed under the shaft; by means of a strut under the knife edge on the end of the lever the correct weight of the initial load is then obtained by the use of platform scales.

17 The largest dynamometer built at present consists of four 60-in. discs and has a power-absorbing capacity of 1500 h.p. at 100 r.p.m., or about 3000 h.p. at 200 r.p.m.

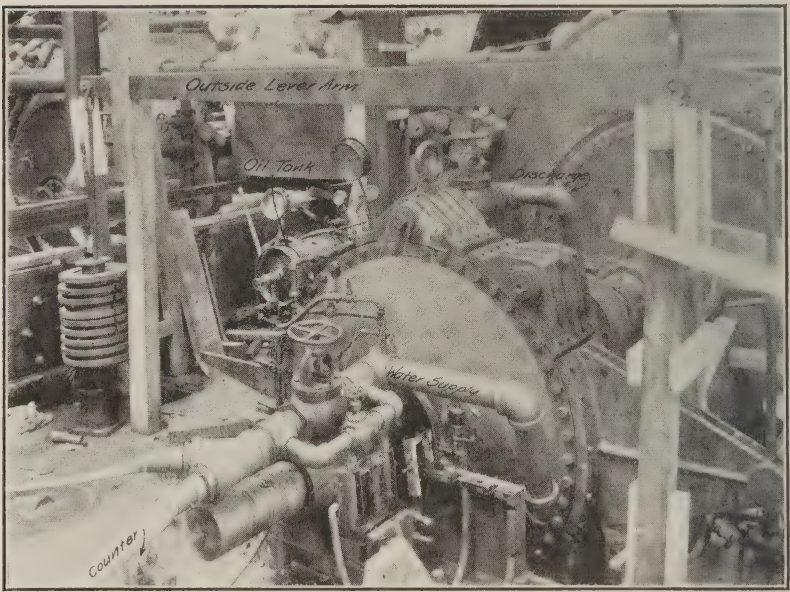

Fig. 6 Dynamometer Rigged for Testing

18 The capacity of the dynamometer is limited by the amount of heat that can be transmitted through the copper plates. This depends upon the range of temperatures and the amount of the circulating cooling water. The capacity is also affected by the kind of lubricating oil used. A cheap grade of cylinder oil has been found satisfactory. A system of forced lubrication is essential to smooth operation.

19 A series of tests has recently been made at the laboratories of the Worcester Polytechnic Institute to determine the relative heat-transmitting properties of copper sheets just as they are received from the rolling mill, and similar sheets electro-copper-plated. These tests were made with a view to increasing the capacity of the dynamometer. The apparatus used consisted of two double-disc Alden dynamometers with the rotating cast-iron discs removed. (See Fig. 8). The dynamometers as tested consisted of an outside cast-iron casing and four copper sheets, with the necessary spacing rings. The dynamometers were identical in every way, except that in one the copper sheets were electro-copper-plated.

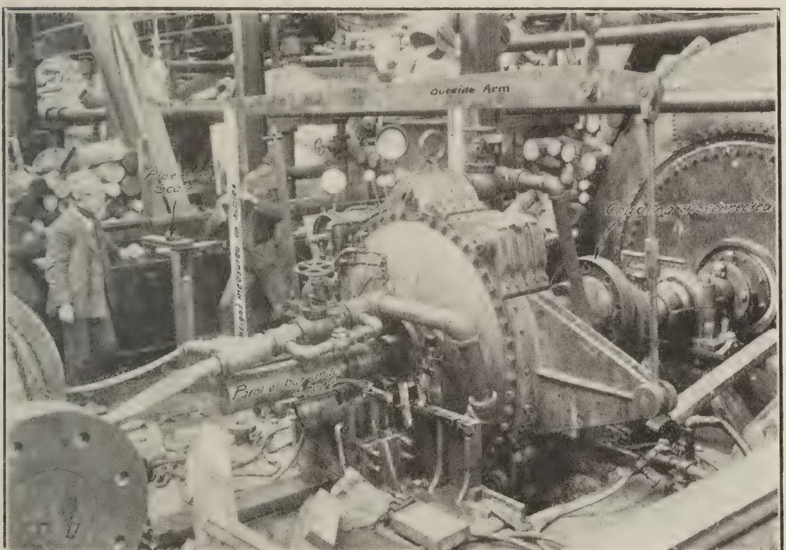

Fig. 7 Method of Calibration After Tests

COUPLING DISCONNECTED, SHAFT RESTING ON PARALLEL BARS, UNBALANCED LOAD OF DYNAMOMETER WEIGHED ON SCALES.

20 The dynamometers were set up so that the spaces normally occupied by the revolving discs were piped to the steam main. Plugs were removed from the top and bottom of these spaces so that all air and condensed steam would be removed. Circulating water was supplied at the bottom of the casings and taken out at the top, both the circulating water and condensed steam being collected and weighed. Thermometers were inserted in the steam line next to the dynamometers and in the water supply line, also in the discharge of

the circulating water and of the condensed steam. Several tests were run on each, of five minutes' duration. The accompanying table gives results of these tests.

	Plain Copper Sheets	Electro-plated Copper Sheets
Average temperature of entering circulating water.	44 deg. fahr.	44 deg. fahr.
Average temperature of exhaust circulating water.	100 deg. fahr.	100 deg. fahr.
B. t. u. per sq. ft. per min. per 50 deg. difference in temperature of circulating water.......	467	610
Increase, in percentage..........................	...	30.8

21 The results were all reduced to a common basis, namely: the B.t.u. transmitted through 1 sq. ft. of copper sheet per min. per 50 deg. difference in temperature of circulating water. The dynamometer containing the copper-plated sheets showed an increased heat transmission of more than 30 per cent over the untreated. A probable explanation of this phenomenon is that when the sheets are electro-copper-plated, the copper is deposited in small globules and the actual surface not only is increased but is made rougher; this tends to mix up the water currents, bringing more new water in contact with the copper.

22 In the actual operation of these dynamometers, the heat is generated on a thin film of oil directly against one side of the copper and the water passes over the surface on the other side, carrying off the heat generated. It is a well known fact that more heat is transmitted through copper than can be readily carried off by the water, and any increase in surface in contact with the water gives a corresponding increase in capacity. As the capacity of these dynamometers depends upon the heat-transmitting power of the copper sheets, it is clear that this capacity can be increased 30 per cent by the use of electro-plated sheets.

23 Owing to the system of continuous forced lubrication, the dynamometers are capable of holding their maximum load for any length of time. A dynamometer recently used held a load of from 2000 to 2300 h.p. during a series of tests on a pair of turbines of over eight hours' continuous running. The reason for making so long a run was that the weir for measuring the water used was situated in the canal above the turbines, and considerable time was required to allow conditions to become constant. It may be of interest to note that this weir was standard, with end contractions, and was

73 ft. long. During the tests, when the wheel gates were wide open, the quantity used by the wheels required a head of 2 ft. on the crest of this weir. Long runs are also required when the current meter is used in measuring the discharge from the wheels.

24 The largest power ever absorbed at one time by these dynamometers was 4100 h.p., developed by a pair of turbines under a head of 110 ft. at a speed of 225 r.p.m. These turbines were used to furnish power for a paper pulp mill. There were six grinders on either side of the turbines, making twelve in all. The grindstones nearest the turbines were removed, and two dynamometers put in their places.

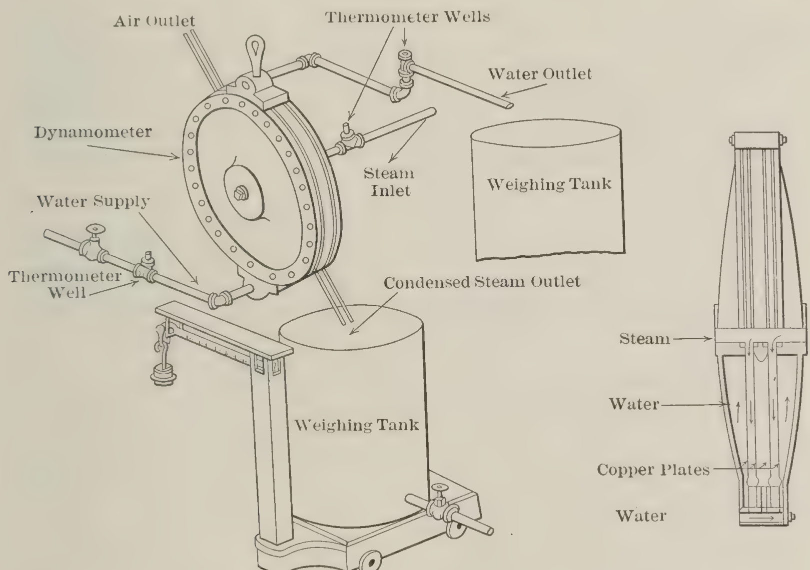

FIG. 8 LAYOUT OF APPARATUS; WITH SECTION OF DYNAMOMETER

25 The amount of work required to make such tests is comparatively small, when the amount of power measured is considered. Two units of approximately 4000 h.p. were tested inside of three days.

26 In pulp mills it is difficult to know how much power the wheels are developing. It is not always safe to base the estimate on quantity of pulp, as there are many variations in the conditions of stones, wood, etc. Hence quite a large percentage of brake testing of wheels has been done in grinder rooms.

27 Several tests of large powers have been made in hydro-electric stations in which case the generators are set aside and the dynamometer mounted on the wheel shaft, usually in place of the half coupling.

28 To give an idea of the actual running conditions sometimes found, a test in one plant showed that the wheels were giving the power called for by the contract, but that if they had been run at 200 instead of 225 r.p.m., the power would have been increased from 2000 to 2300 h.p.; thus showing a waste of 300 h.p. and a correspondingly lessened efficiency. On another test it was found, that when the wheels were running at the guaranteed speed, the power and efficiency were between 30 per cent and 40 per cent lower than they should have been. Removing the entire load only slightly increased the speed.

29 Many tests have been made which show that if the wheels are given a "fair setting," and the speed properly chosen, they will agree closely with the computed ratings made from the tests at the Holyoke flume. One test of a pair of wheels, made after installation, agreed so closely with the computed results from the Holyoke tests of the individual wheels, that the curves showing the relation of the horsepower and efficiency to the speed-gate opening of the pair came between the curves of the separate wheels transferred to the same head basis. The term "fair setting" means that the water should be brought to the wheel with a low velocity; the draft tube should be designed especially for the particular conditions, and should be air-tight; the wheels, if a pair, and center-discharging, should not be set too closely together; if a pair and outward-discharging, end supply should be avoided if a steel penstock is used; the shaft of the wheel should not be larger than necessary; the bearings should be kept in line, etc. The setting just described refers to open flumes or steel penstocks and not to spiral casings.

30 Incidental to the brake testing of water wheels, considerable information has been obtained regarding the efficiency of large bevel gears, such as are commonly used in transmitting power from vertical wheels in low-head installations. About a year ago, a series of tests were made to determine: first, the horsepower delivered to the horizontal generator shaft from two vertical wheels; second, the horse power of the individual vertical wheels. By subtracting the former from the sum of the latter, the loss due to the bevel gears and bearings was obtained. The tests were conducted in the above order so as to get the ouptut at the generator coupling with gears running normally. Then the gears were removed and the individual vertical tests made.

31 The total horsepower delivered to the generator was approximately 700. (See Fig. 9 and Fig. 10). The driving gear was of the

ordinary wood-mortise type, outside diameter 6 ft. 5 in. approximately, with 68 teeth, 14 in. wide, meshing with a cast-iron pinion which had 48 teeth with a planed tooth outline. At full load the loss in the gears was 3.5 per cent and 3.4 per cent for two separate units, or the efficiency of the horizontal-shaft vertical-wheel gear drive was about 96.5 per cent. The gears were well lubricated with a thick grease.

32 About nine months later it was necessary to test one of these same units in exactly the same manner. The loss in gears this time was a trifle less, the tests giving 3.1 per cent. As a matter of fact,

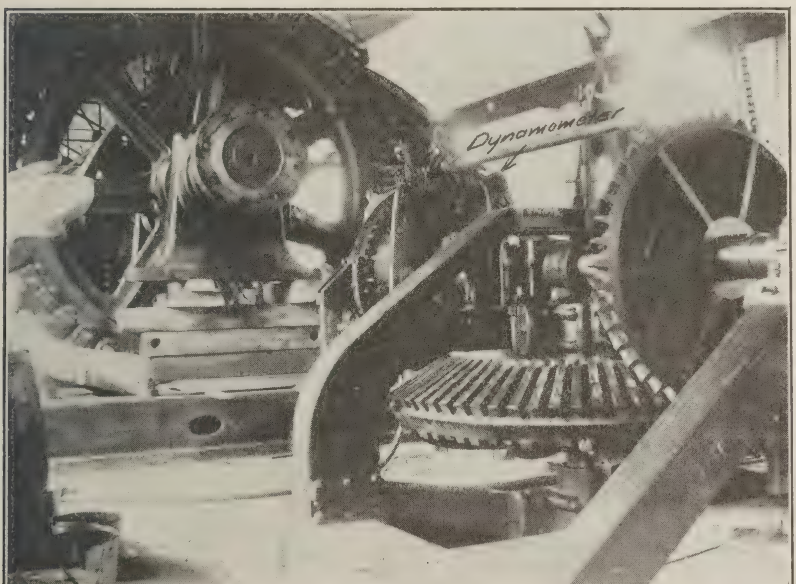

Fig. 9 Horizontal Testing of Vertical Wheels with Bevel Gears

the gears were running smoother at the last test, having had nine months more of service. When the first tests were made, the plant had been running less than a year.

33 Besides this direct measurement of gear efficiencies, two separate tests have been made within the past year, of two vertical Boyden wheels with the power measurements made on the horizontal shaft. These wheels had been in operation for over thirty years. The bevel gears (both the driving and the driven) were of cast iron in each unit. No brake tests were made on the vertical-wheel shafts but

the discharge from each was carefully measured over a standard weir by Mr. A. F. Sickman, hydraulic engineer for the Holyoke Water Power Company. The best efficiency of the wheels at full gate, which in this case includes the loss of gears and bearings, was 83.5 per cent and 83.7 per cent, respectively.

34 Several similar tests have been made with corresponding evidence as to good efficiency of the gears. In one plant, containing seven pairs of vertical wheels, where all the mortised wooden gears and cast-iron pinions had been in use for over fifteen years these were still good, without having had even a tooth changed.

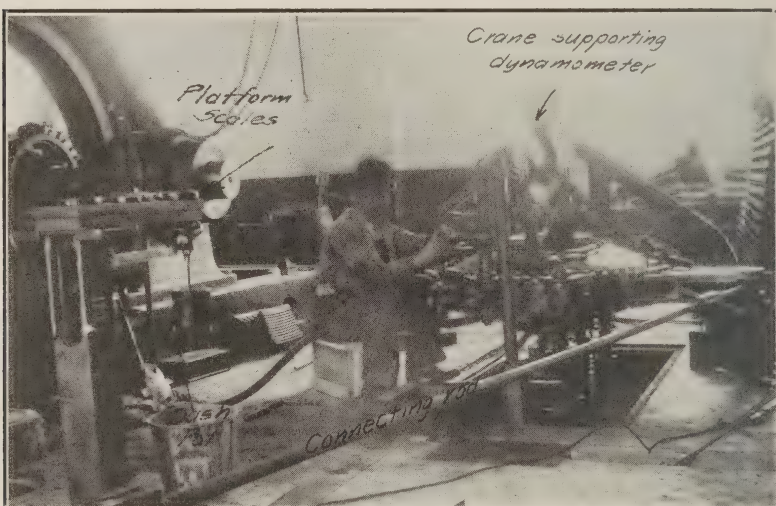

Fig. 10 Testing of Vertical Wheel (Gear Removed)

35 All of the information obtained concerning the loss due to bevel gear drives, leads the writer to conclude that if gears are properly designed, set up, and operated, and are not overloaded intermittently or continuously or left to care for themselves, they should show an efficiency of from 95 to 97 per cent.

36 Figs. 11 and 12 show two settings in which the wheels were tested after installation. The results of the tests, together with the results of the tests on the same wheels at the Holyoke testing flume, and reduced to a common head-basis, are given in the curves, Figs. 13 to 16 inclusive.

37 The setting shown in Fig. 11 is for a pair of 36-in. wheels operating under a head of 28 ft. at a speed of 200 r.p.m. The wheels are

set 3.3 diameters apart and the draft tube changes from a circular to an oval cross section with a constantly increasing area, and discharges horizontally into tail-race. In this particular installation, the velocity of the departing water from the draft tube was found to increase materially the working head on the wheels. The results of the brake tests made after installation, as shown by the accompanying curves, checked with the Holyoke tests computed for the same head at full gate. It may also be of interest to note that the generator tests made immediately following also checked with the brake tests. The difference between the two tests as shown in the curves at part gate openings is probably due to different methods of setting the gates under test. The full gate opening, however, was exactly alike in both cases.

38 In the setting shown in Fig. 12 the wheels actually give a considerable increase in power over that computed from the Holyoke tests. This is probably due to two reasons, the first and most important being that the Holyoke testing flume is not designed to test wheels of this size; and second, that the setting is exceptionally good. The wheels are 48 in. in diameter and are set 4.25 diameters apart with ample space around the wheels in the casing and with well-proportioned center case and draft tube. The curves in Fig. 15 and Fig. 16, representing the results of tests made on these wheels both at Holyoke and after installation, show that the best speed at full gate as computed from the Holyoke tests is about 182 r.p.m., and the best speed after installation 195 r.p.m. Comparison of the horsepowers at best speed shows an increase after installation of 40 h.p., but if the horsepowers at 195 r.p.m. be compared, then there is an increase of 130 h.p. This apparently excessive increase in power is to a large extent due to the fact that the best speed after installation was 195 r.p.m., while from the curve of the Holyoke test the power at 195 r.p.m. has materially dropped off.

39 The curves in Fig. 17 and Fig. 18 show the results of tests on a pair of 45-in. wheels under a head of 44 ft., with a normal speed of 200 r.p.m. It is seen that the discharge increases with the gate opening but the power begins to drop off at 0.75 gate opening and remains contant from 0.9 to full, which accounts for the efficiency curve dropping off so rapidly. In this installation the wheels were set too closely together (less than three diameters apart) and the shaft was excessively large. In accordance with data obtained from our recent tests these wheels should have been set at least four diameters apart.

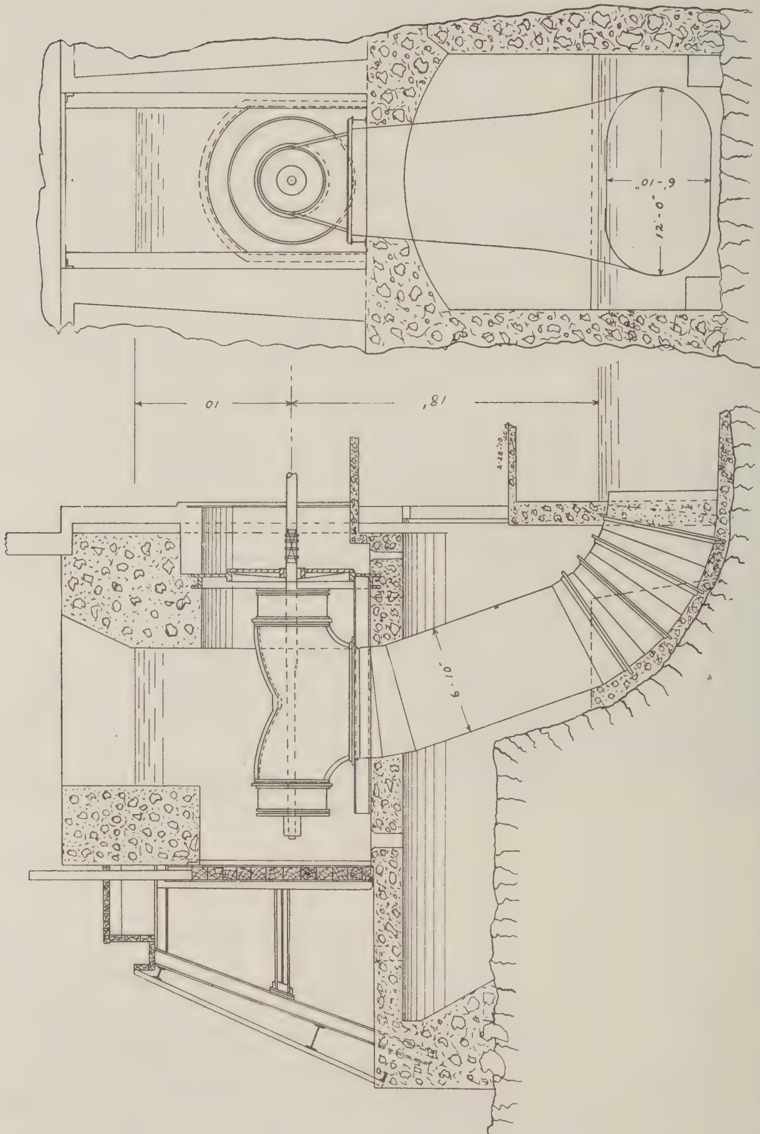

FIG. 11 SETTING OF A PAIR OF 36-IN. WHEELS UNDER 28-FT. HEAD. DISTANCE BETWEEN WHEELS 3.3 DIAMETERS

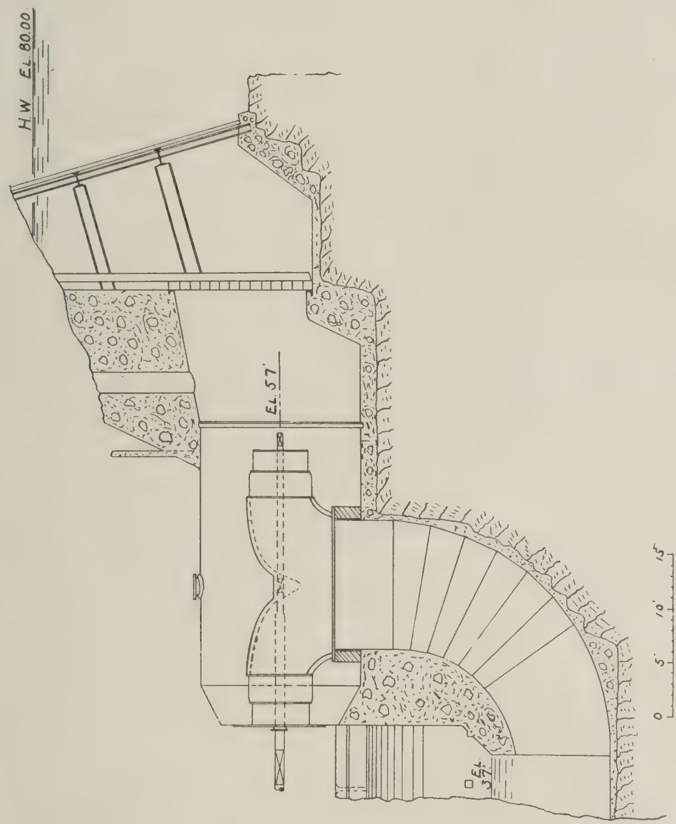

FIG. 12 SETTING OF A PAIR OF 48-IN. WHEELS UNDER 40-FT. HEAD. DISTANCE BETWEEN WHEELS 4.25 DIAMETERS

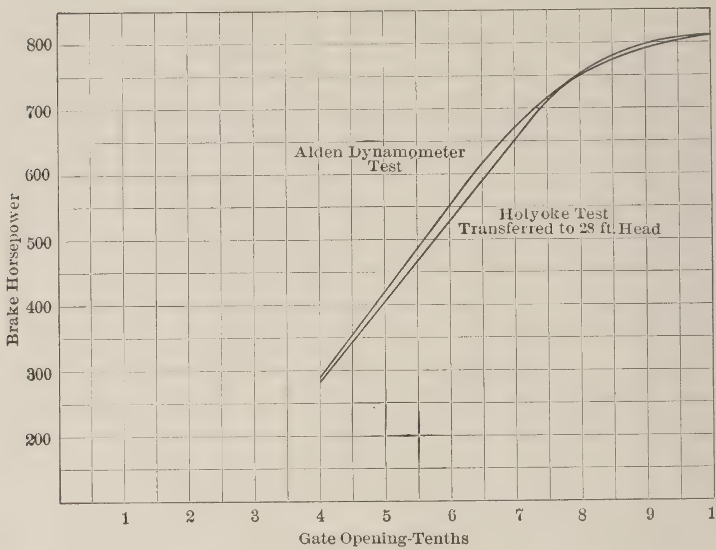

FIG. 13 CURVES SHOWING RELATION BETWEEN HORSEPOWER AND GATE OPENING FOR A PAIR OF 36-IN. WHEELS UNDER 28-FT. HEAD AND 200 R.P.M.

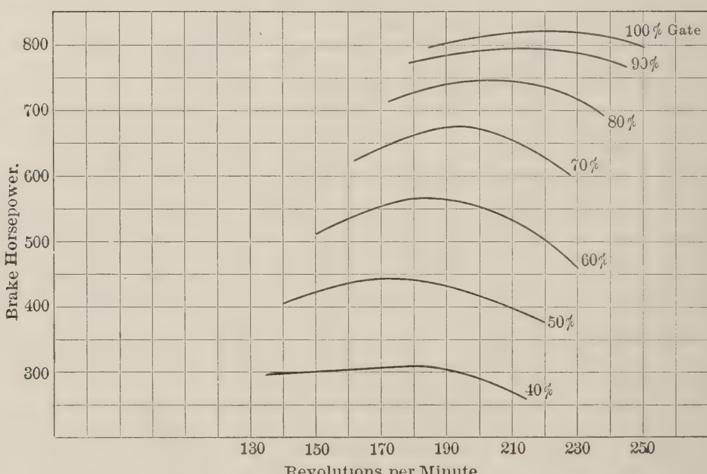

FIG. 14 CURVES SHOWING RELATION BETWEEN HORSEPOWER AND SPEED FOR A PAIR OF 36-IN. WHEELS UNDER 28-FT. HEAD

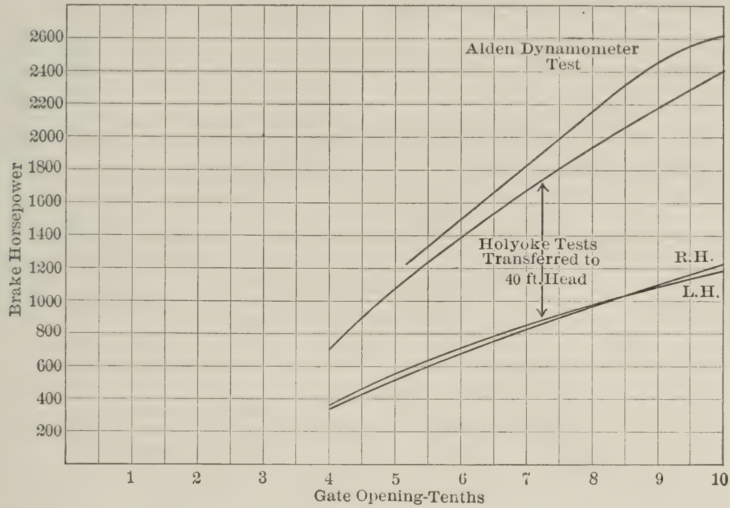

Fig. 15 Curves Showing Relation Between Horsepower and Gate Opening for a Pair of 48-in. Wheels under 40-ft. Head and 200 r.p.m.

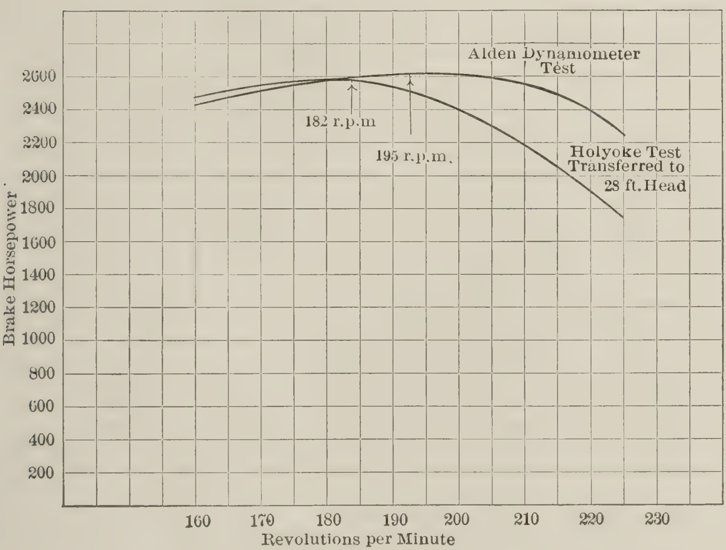

Fig. 16 Curves Showing Relation Between Horsepower and Speed at Full Gate For a Pair of 48-in. Wheels under 40-ft. Head.

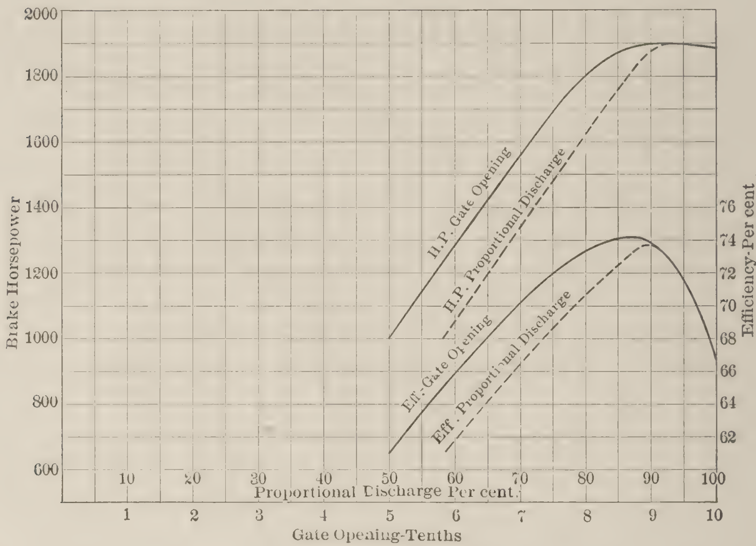

FIG. 17 CURVES SHOWING RELATION BETWEEN HORSEPOWER AND GATE OPENING, HORSEPOWER AND PROPORTIONAL DISCHARGE, EFFICIENCY AND GATE OPENING, EFFICIENCY AND PROPORTIONAL DISCHARGE; PAIR OF 45-IN. WHEELS, 44-FT. HEAD AND 200 R.P.M.

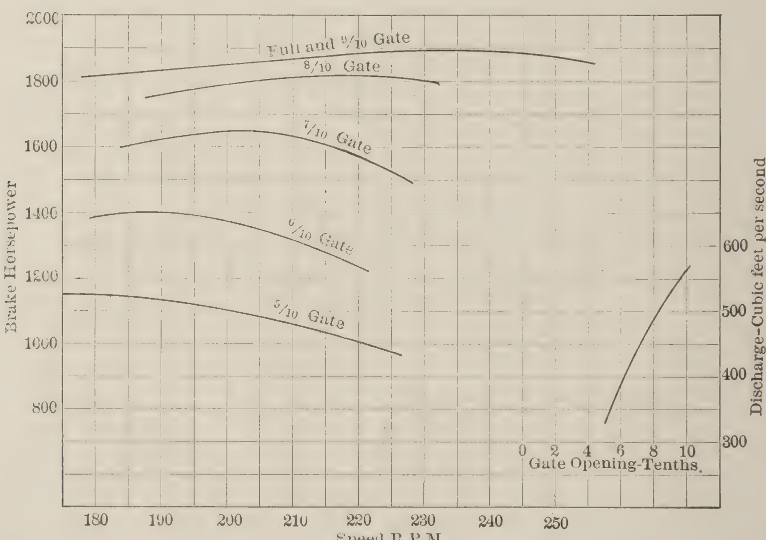

FIG. 18 CURVES SHOWING RELATION BETWEEN HORSEPOWER AND SPEED, GATE OPENING AND DISCHARGE; PAIR OF 45-IN. WHEELS, 44-FT. HEAD

40 While the tests on the 48-in. wheels shown in Fig. 12 are the best example that we have of large turbines actually showing more power when tested after installation than that computed from the Holyoke tests, it is by no means the only instance of such results. Within the past five years there have been made over 100 complete brake tests of wheels at different plants in all of the New England States and most of the Eastern Atlantic States. Most of the leading makes of wheels have been tested in this manner, and under heads varying from 10 ft. to 200 ft. Information derived from this experience confirms the following general statement:

41 That the performance of low-head turbines of diameters up to about 36 in., as computed from the Holyoke tests, should be attained after installation; that turbines greater than 36 in. in diameter should show better results after installation than those computed from the Holyoke tests, the amount of difference increasing with the diameter of the turbine.

42 There is still a vast deal of information needed concerning the behavior and proper settings of wheels after installation, before the subject can be put upon a good working basis, and it is hoped that enough points have been touched upon in this paper to call forth a goodly amount of discussion and reliable information that may be of value.

MECHANICAL FEATURES OF ELECTRIC DRIVING IN MACHINE SHOPS

By John Riddell, Schenectady, N. Y.
Member of the Society

It is with the mechanical features of motor driving that this paper is to deal, and chiefly with what has been done in the electrical equipment of the most commonly used machine tools in the plant of the General Electric Company, at Schenectady, with a few sketches of some large work that has been erected outside.

2 Some ten or twelve years ago it was decided to erect a large and up-to-date electrically driven machine shop, and plans were started some time ahead of the completion of the building. At first the plan was to have every machine tool individually driven, but the time was so short that we abandoned this idea and concluded to arrange the machines in groups, driven by a motor direct-coupled to the end of a section of lineshaft. This arrangement was used only to take care of small and medium-sized machines, of which few, if any, were at that time equipped with individual motors.

3 Considerable difficulty was experienced in arranging the lineshafts and countershafts in this system, owing to their being traversed by small side-bay electric cranes. Fig. 1 gives a general idea of this method of group-driving. However crude it may appear in the light of present practice, it was considered in its time thoroughly up-to-date.

4 One of the mechanical difficulties encountered in attaching individual motors to small machines was the unwieldy size of some of the earlier motors of small capacity. Sometimes the motor would be as large or larger than the machine and this feature was largely responsible for the prevalence of group-driving of small machines, even where the individual drive would have been preferred. Recent improvement in motor design has led to a great reduction in the size of motors for a given capacity, so that the 25-h.p. motor of today is not nearly so

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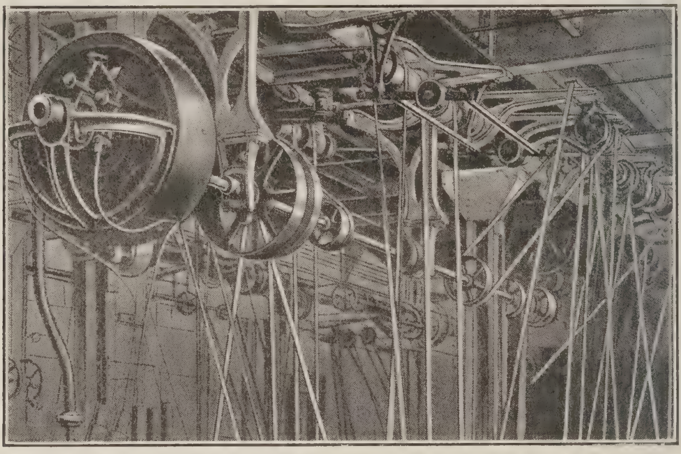

Fig. 1 Method of Group-Driving

large as the 10-h.p. motor of earlier years. It is therefore much easier now to make the motor an integral part of the machine; and even where only fractional parts of a horsepower are required, for light operations,

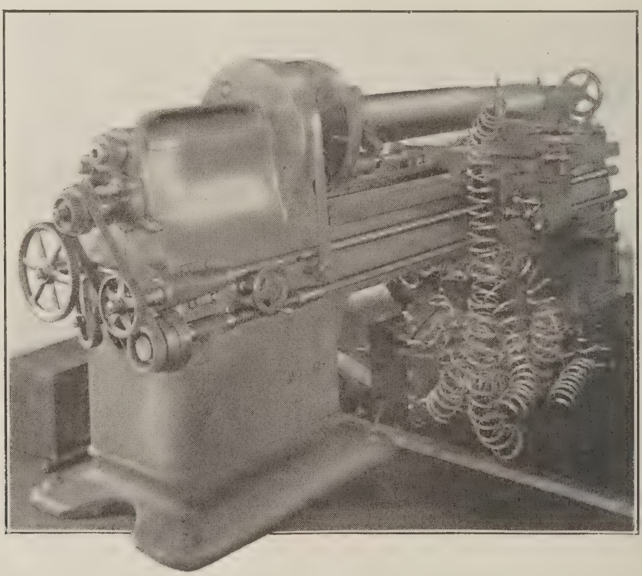

Fig. 2 Lathe Driven by Induction Motor Concealed in Cabinet Leg

suitable motors of very small weight are now available which are well adapted in size to the smallest tools.

SMALL MACHINE TOOLS

5 Figs. 2, 3 and 4 show good examples of individual motor drives, in which the motors are inconspicuous and form integral parts of the machine tools. Fig. 2 shows a lathe driven by an induction motor concealed in the cabinet leg. Fig. 3 illustrates a wood-boring machine driven by an induction motor through a single pair of bevel gears. The inverted motor is bolted to a plate, which is in turn bolted to the

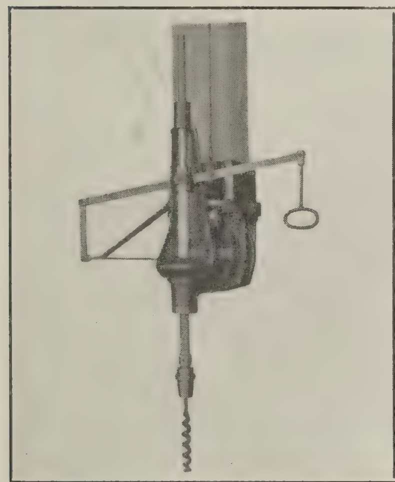

Fig. 3 Wood-Boring Machine Driven by Induction Motor, Through Single Pair of Bevel Gears

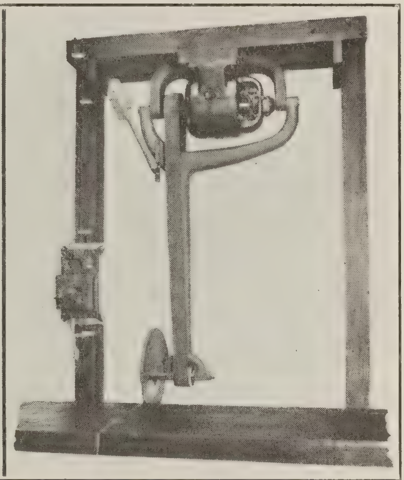

Fig. 4 24-in. Reliance Swing Saw with Motor

bottom of a post and to the frame of the tool. At first, the plate was arranged to swivel on the tool post, in order to provide means for moving the tool longitudinally over the work; but later, this adjustment was abandoned, as it proved to be easier to move the work horizontally with reference to the tool. This tool is a good example of the compactness of the electric motor, and its easy adaptability to wood-working machinery. Fig. 4 shows the application of a direct-current motor to a 24-in. swing saw.

LARGE MACHINE TOOLS

6 The large lathes, vertical boring mills, planers, milling machines, etc., were supplied each with its own individual motor, which was a marked improvement over the original belt-and-countershaft methods of driving. Since starting this work, the company has changed over several thousand machine tools to motor drives. Most of the difficulties in changing from belt to motor driving were in making suitable

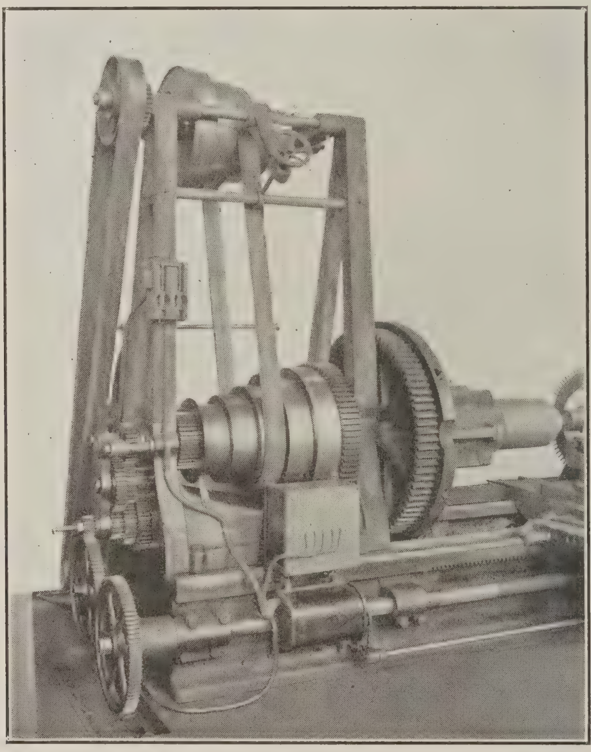

FIG. 5 42-IN. MILES-BEMENT LATHE DRIVEN BY CONSTANT-SPEED MOTOR' USING CONES AND BELTS

connections between the motor and the tool to be driven. We do not pretend that in every case we have adopted the best arrangement, as in many cases the machine tool is not of sufficient value to warrant an expensive mechanical connection.

7 Take, for example, a medium-sized lathe of relatively moderate value. If an expensive transmission device was required in order to apply a motor to the lathe, the total cost of the lathe, as changed, might easily be more than the price of a new lathe especially designed for motor driving. In such cases it is found expedient to erect the countershaft and cone about four feet above the headstock, on suitable brackets, fasten the motor in a convenient place on the machine, and continue the use of cones and belts. Such an arrangement is shown in Fig. 5.

8 In all cases where old machine tools are converted to electric driving, it is desirable to mount the motor on some part of the machine

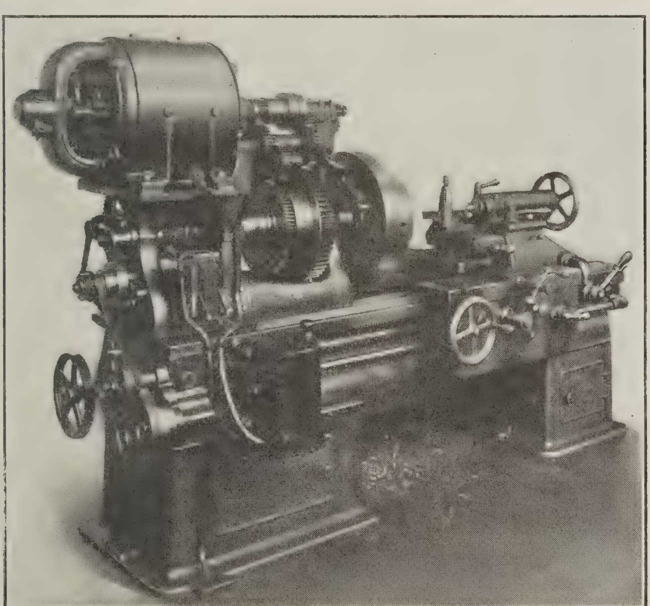

Fig. 6 24-in. Fitchburg Lathe Driven by Motor, with Reversing Gears

if possible, rather than on the floor near the machine. In the former case, the machine tool constitutes an independent self-contained unit which can be moved by the crane as a whole and located wherever desired. Cleanliness is promoted, by leaving a clear floor space to sweep, and the motor is less liable to accumulations of dirt caused by sweeping. There is also a tendency for the motor and the machine tool to become shifted out of alignment, if they are separately mounted on the floor.

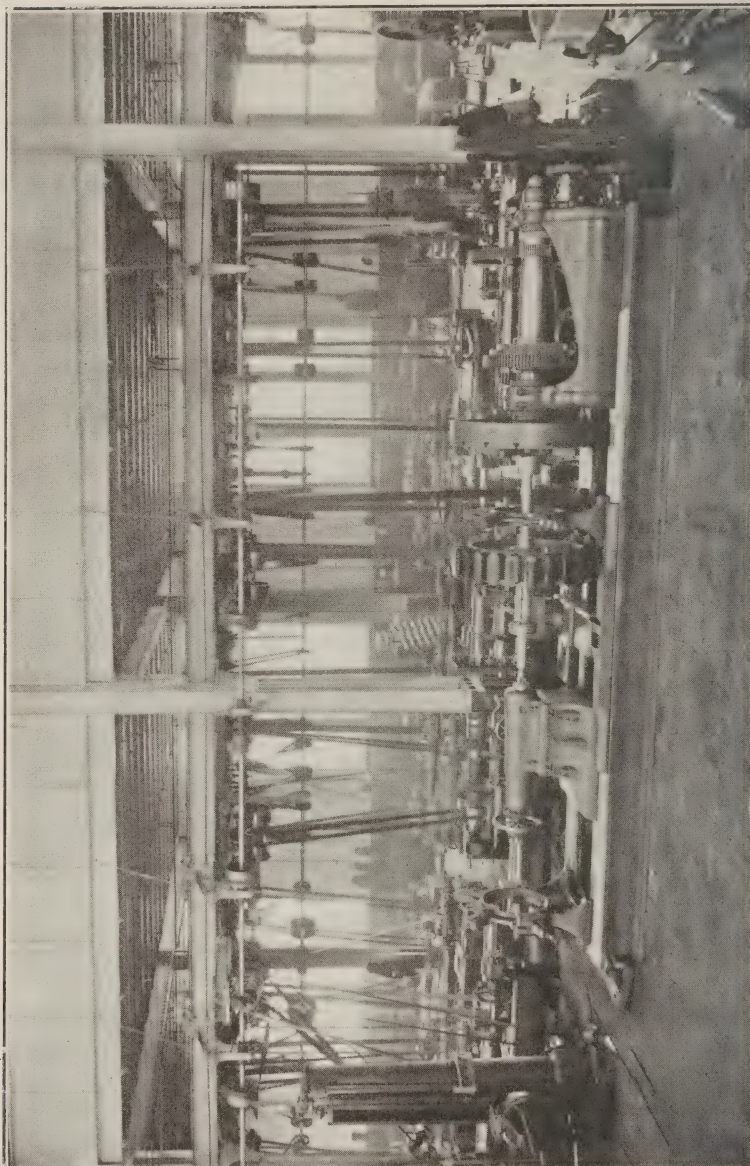

FIG. 7 LATHE, WITH MOTOR IN HEADSTOCK BENEATH SPINDLE

9 For lathes of more importance, and where the value of the tool warrants, we make an all-gear drive, with reversing gears, which are used principally for screw cutting. Fig. 6 shows a lathe equipped with reversing gears. More recently we have produced motors which can be very quickly reversed, obviating the necessity of reversing-gears.

10 When the original change was made from belt to motor drive there was one particular triple-geared lathe, 72-in. swing, to which we applied a two-to-one variable speed motor. The only change necessary in this case was to substitute two gears for the lathe cone, mounted on a quill, and made to engage with a pair of rocking gears on the motor. This gave a speed variation of four to one at the motor, and with the triple gears of the lathe, we had an exceedingly fine speed range. This lathe is shown in the foreground of Fig. 7. The motor is

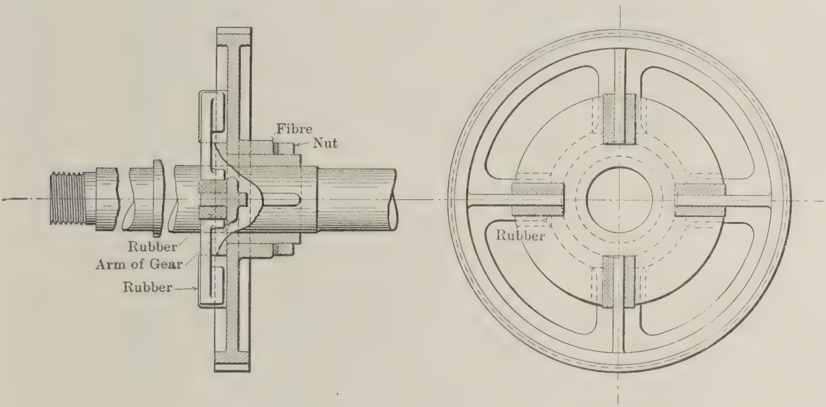

Fig. 8 Diagram of Device to Obviate Chatter Marks on Finished Work

placed in a most advantageous position, in the headstock underneath the spindle. During about ten years' service it has never been removed.

11 In a shafting department like that of the General Electric Company, where the range of size does not vary over three or four inches on the standard work, no very great speed changes are necessary, and a two-to-one motor usually has range enough to meet all requirements. What is particularly needed is ample power, strength of parts, and simplicity of construction, especially in lathes used for roughing, which are usually handled by unskilled labor.

12 For finishing shafts, however, where greater accuracy is required, an all-gear drive with steel gears is not satisfactory, because the chattering set up by the action of the gear teeth is very apt to be trans-

mitted to the finished work, leaving parallel ridges. This difficulty was overcome in some special lathes which we had built for the purpose, in which the driving gear on the main spindle was left loose and acted on the driving plate, keyed to the spindle, through four rubber buffers. This device is shown in Fig. 8.

13 More recently, trouble from this same cause was experienced in one of our tool departments, and was corrected by the use of a pinion made of muslin. This pinion has several features which specially adapt it to motor-driven machine tools. It is practically noiseless and

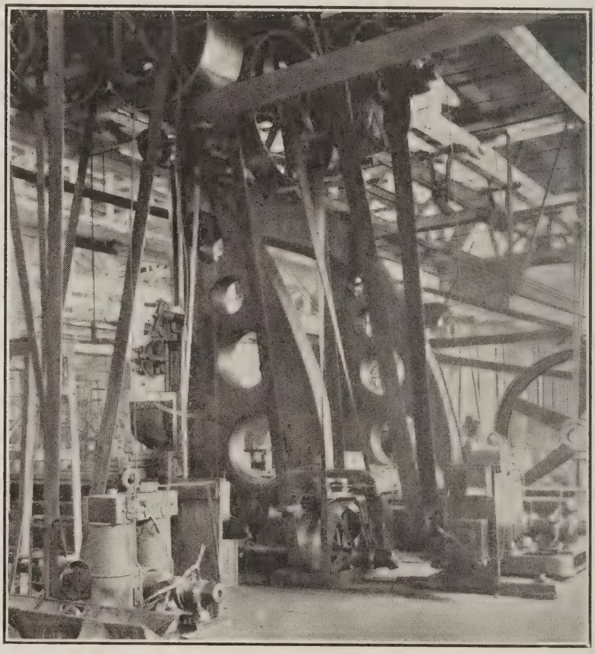

Fig. 9 Evolution of Application of Motors to Planers
LINESHAFT DISCARDED. COUNTERSHAFT DRIVEN FROM MOTOR ON FLOOR

very durable, does not shrink, and is sufficiently flexible and elastic to absorb vibrations which might be transmitted to the finished work.

14 Another application to motor driving in connection with lathes, and one that has been much appreciated, is the use of an auxiliary motor to operate lathe carriages having a long travel, of about 35 ft. The motor is bolted to one side of the carriage, and carries a pinion which meshes with the hand-wheel gear. A two-way switch is pro-

vided for operating the motor in either direction. The use of a motor not only saves the operator a difficult task, but it has also proved a great economy of time. It formerly required 30 to 35 minutes to shift the carriage through its full travel, and the hand wheels were placed so low that a man had to stoop to use them. The motor will move the carriage from one end of the lathe to the other in a minute and a half, and it can be stopped at any point within a 1/16 in. of the cut.

15 The best location for a motor on a lathe, and on most machine tools, is as low down on the machine as possible. The amplitude of the vibrations set up will be smaller, the closer the motor is to the floor, and the liability of chattering will therefore be reduced. The location of the motor in the cabinet leg, or in the headstock of a lathe, as shown in Fig. 2 and Fig. 7, is ideal, but there are, of course, many cases where the motor must be mounted over the headstock because no other place is available. The necessity of having the motor out of the way of the work is obvious, as turnings of chips, if allowed to get into the motor, would at once give rise to electrical troubles, especially in direct-current machines.

CONTROLLERS

16 The location and arrangement of controllers for lathes depend upon the class of work to be performed. Where the lathe is started, stopped, and varied in speed by the controller, the latter should be mounted on the front of the lathe, and the handle extended by means of a shaft to the lathe carriage, where it will be constantly under the hand of the operator. Ease of control unquestionably results in the rapid and economical production of work. Where the work varies considerably in diameter, frequent changes of speed will be required, and where the most efficient cutting speed can be obtained by simply turning a conveniently located handle, the work will be turned out at a maximum speed. If frequent shifting of belts is required, a great deal of the work will be done at less than maximum speed, owing to the extra exertion involved.

17 For lathes with constant-speed motors, operated with clutches and shifting levers, or machines on which continuous automatic operations are carried on, such as screw machines, the motor can be kept running for long periods without attention from the operator. In such cases the controller may be mounted at any convenient place on the machine, or near by on the floor, by means of a bracket.

PLANERS

18 Figs. 9, 10 and 11 illustrate the evolution of the application of motors to driving planers at the Schenectady shops. The first step, as shown in Fig. 9, was simply to discard the lineshaft and drive the countershaft from a motor placed on the floor. When this planer was operated by belts it was next to impossible to reverse it in a shorter space than about thirty inches, and even then with a great deal of wear and tear on the belts. Early in 1900 the company produced their first magnetic clutches for driving planers. The first of these clutches was applied to a Bement-Miles planer, 10 ft. wide by 20 ft. long. With this clutch, Fig. 10, we are able to reverse the planer prac-

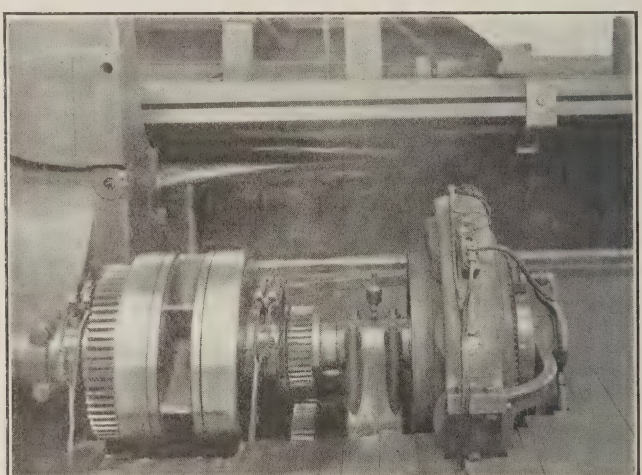

Fig. 10 Evolution of Application of Motors to Planers
MAGNETIC CLUTCH

tically to a line, and to reduce the space required for reversing to about 12 inches. Some trouble was experienced, however, with these first magnetic clutches, owing to the design of the magnets, and pneumatic clutches of a peculiar design were subsequently adopted and have been entirely satisfactory. Fig. 11 shows the arrangement of the motor and pneumatic clutch, as applied to the planer.

19 Our second lot of magnetic clutches was redesigned to eliminate the difficulties experienced with the first lot, and the new ones were applied to a number of portable slotters. These machines have been in continuous operation practically night and day up to the present

time. The clutches have operated with entire success, and I believe the magnetic clutch will eventually be found an important and efficient feature of transmission gears for planers and slotters.

VERTICAL BORING MILLS

20 In our original scheme for attaching motors direct to boring mills, an all-gear drive, with variable-speed motors, was selected, and with very slight changes, has been employed up to the present time. Fig. 12 shows a number of boring mills equipped in this way; the motors

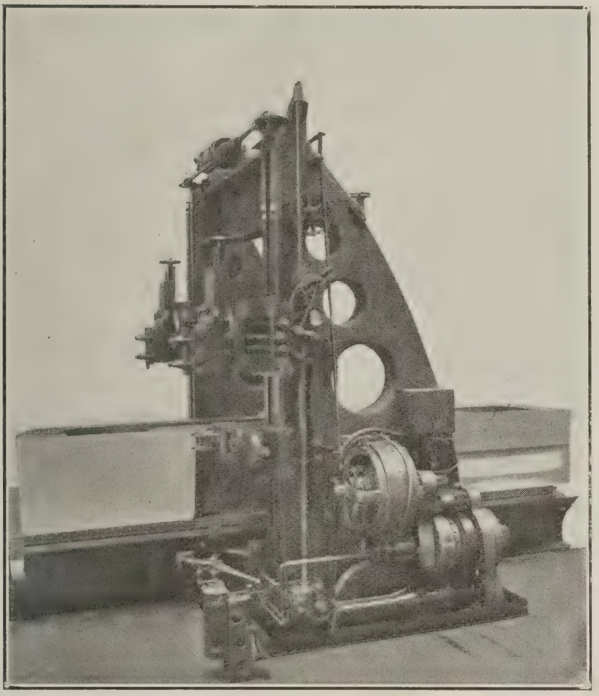

Fig. 11 Evolution of Application of Motors to Planers

COMPRESSED-AIR CLUTCH, TWO CONES

not being visible in the illustration because they are placed below the floor level between the side frames back of the revolving table. A solid foundation is laid, extending under the entire machine, a depression in the back between the side frames forming a bed for the motor, which is securely fixed in position. This common foundation makes the motor and the machine a single compact unit, and no additional

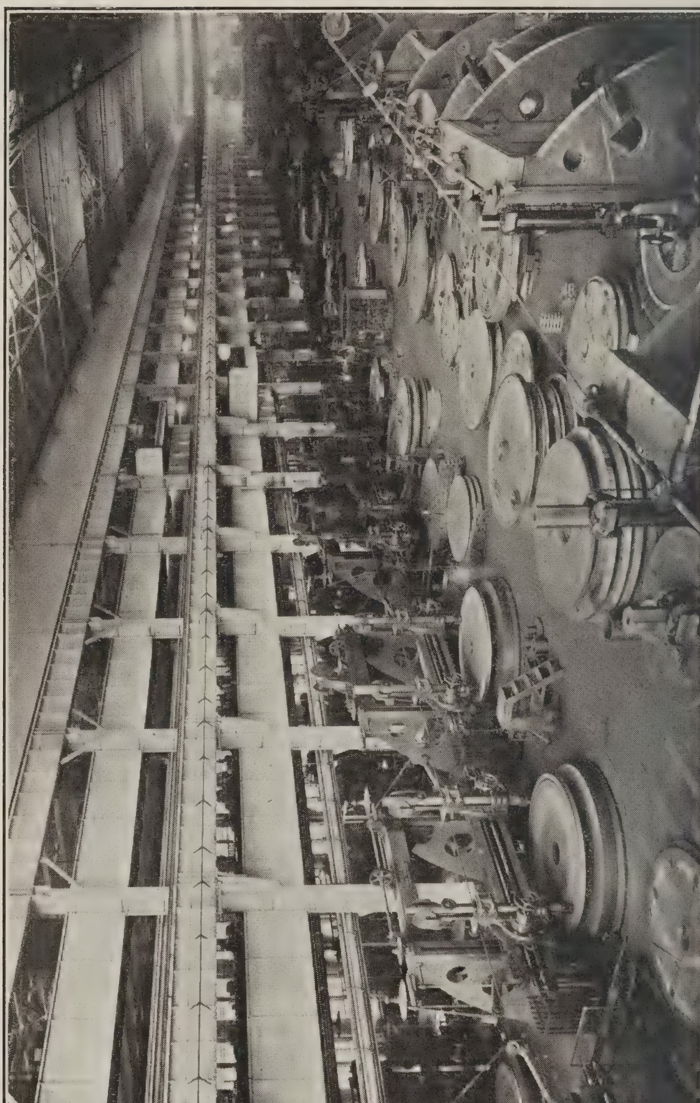

Fig. 12 Interior View of Building No. 60

floor space is required for the motor. No work put upon the table can interfere with the motor, the gears are entirely out of sight, and the controller is placed at the right-hand side of the machine, where the operator usually stands.

21 On boring mills from 20 ft. to 25 ft. in diameter, with the usual slow intermediate and direct-gear drive that comes with the mill, a variable-speed motor of two-to-one ratio gives a very satisfactory speed range.

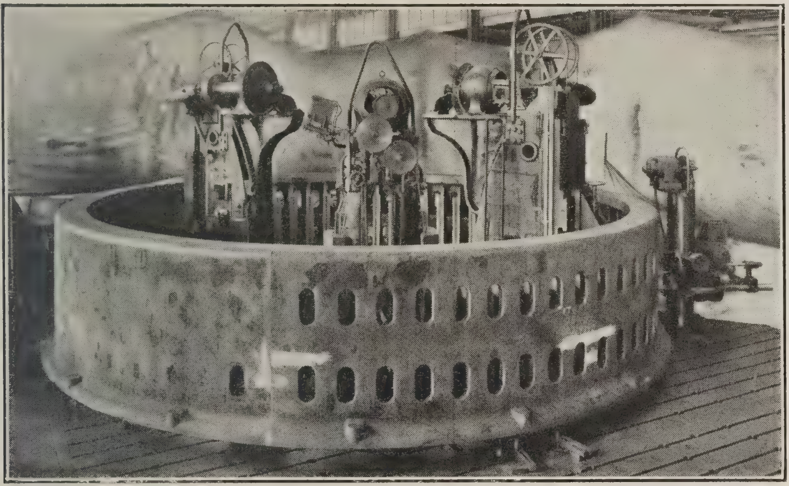

Fig. 13 Application of Motor to Machine Tools on Iron Floor Plate

PORTABLE TOOLS

22 Various machine tools of the portable type are used in ordinary large machine shops, and are placed in various positions on iron floor plates. The efficiency of these machines, such as rotary planers, slotters, etc., used in erecting departments, has been greatly increased by the use of electric motors. A group of these tools working on a large casting is shown in Fig. 13. The use of portable tools was almost impossible before the advent of the electric motor, but now the machine tools used in erecting shops, and in isolated places away from the source of power, when equipped with electric motors are ready to run at a moment's notice. On up-to-date rotary planers the motor is placed on the carriage; the under-side of the bases is planed, and means are provided for transfering the planers by electric cranes.

23 Under the old arrangement, if machinery stood idle for days and weeks, the countershafts and loose pulleys were so neglected that they would squeak; some one would then throw off the belts to stop the noise, and two hours' work was frequently necessary before they could be started for a hurried job. The same thing is true of some boiler makers' and blacksmiths' machines, such as rolls, shears, cutting-off machines, etc.

BELTING

24 Twenty-five or thirty years ago, in the days of the old jobbing shop, the buildings were not so high-studded, and the lineshafts and countershafts were usually within reach of a twelve-foot or fifteen-foot ladder at the most. Cone belts running to machine tools were very easily manipulated, and an expert lathe hand would never think of using a pole for shifting his belt from one step of the cone to another. But in these days of sanitary buildings, with ceilings from twenty to twenty-five feet high, it becomes an exceedingly difficult problem to arrange countershafts within reasonable heights; to say nothing of the necessary length of the vertical belts, the dust set in motion, and the difficulty of painting and whitewashing ceilings for the sake of cleanliness.

25 Another condition of lineshaft driving which has not been much spoken of, of late, is the difficulty of keeping the shaft in alignment, where the hangers are suspended from the roof trusses. The writer has seen such shafts five or six inches out of alignment, due to a heavy fall of snow on the roof, or to the settling of foundations. Another trouble is due to state laws and shop rulings, where a few trained men are employed as belt-lacers, and it is against the rules for men who are not belt-lacers to do the work. This is the cause of numerous delays, with consequent loss of production.

26 The only advantages that may be claimed for belts is that they take up the vibration of the gears, and thus prevent chatter marks on fine work; and that they will slip under over-load and be thrown off the pulley, stalling the machine, instead of breaking the tool or spoiling the work. It has already been explained how the effect of vibration has been remedied by means of rubber buffers or muslin pinions, and there are exceedingly few cases where belt-slip is not a detriment rather than an advantage. The use of high-speed tools calls for a considerable increase in the power necessary to drive the machines, as these tools take a heavier cut at a higher speed than those of car-

bon steel. These conditions make belt-slip very objectionable, and one of the chief advantages of motor driving is that it increases the power of machine tools beyond the capacity of belts of reasonable length. Modern practice is in the direction of eliminating the belt almost entirely, although there are a few machine tools, such as the older types of automatic-screw machines, grinding machines and some wood-working machines, on which they are necessarily retained.

27 The great majority of metal-working machines are best adapted to motor driving through all-gear connections, although a few machines, such as small grinders, buffing wheels, polishing wheels, etc., are best connected direct to the motor shaft.

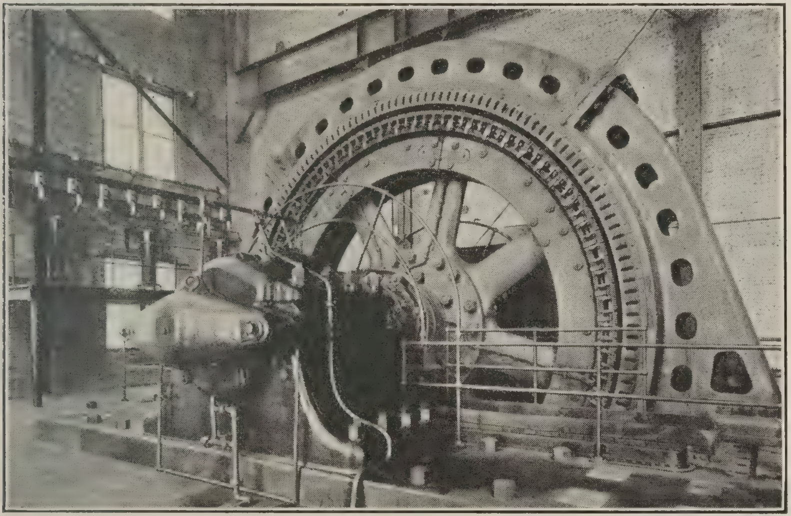

FIG. 14 6000-H.P. INDUCTION MOTOR, RAIL MILL, INDIANA STEEL COMPANY GARY, IND.

STEEL MILL WORK

28 The application of motors to metal-forming machines, such as rolls for rolling steel, charging cranes, etc., does not belong strictly to machine-shop practice, but the work of these machines is closely allied to what is required of other metal-working machines. Some of this work now performed by motors, is the heaviest kind of mechanical work ever attempted by any kind of prime mover. Fig. 14 shows a large induction motor driving an up-to-date steel rail mill, requiring 6000 h.p. or more. The operation of an entire mill is fre-

quently dependent on the continuity of operation of each piece of apparatus. For this reason it has been the practice to build the steam engines which drive mill machinery of the most substantial material and design. In replacing engines by electric motors, these same features were embodied in the motors.

29 The breakable coupling between the engine and the rolls is retained when motors are used. This coupling is of such strength that it breaks before the rolls are injured by shocks. The coupling frequently breaks diagonally, and a very heavy end-thrust is sometimes

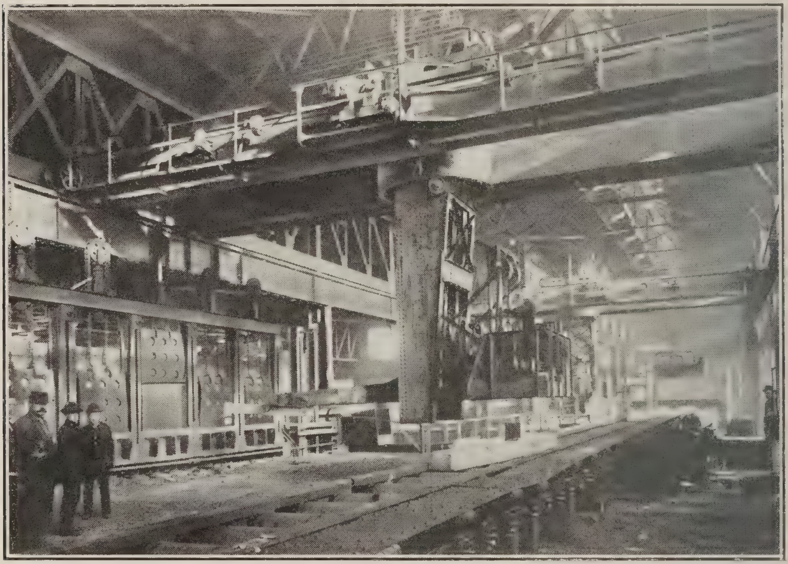

Fig. 15 Charging Crane, Blooming Mill, Illinois Steel Company, South Chicago, Ill.

produced, tending to separate the ends of the coupling. With the engine drive, this end-thrust frequently slides the roll housings out of their places, and considerable time is required to replace them. When motors are used, this difficulty is readily avoided by allowing the shaft of the motor to slide longitudinally in its bearings. To keep the motor shaft in its proper position, a breakable end-thrust bearing has been devised, which allows a bolt to break at a predetermined pressure before any other part of the machinery can be injured. This breakable bearing is shown in Fig. 14. It acts through the breakage of the long bolts shown alongside of the bearing housing.

The ready replacement of the parts when the coupling breaks is only one of the advantages incidental to the use of the motor for driving rolls.

30 Another style of motor, known as the mill motor, is illustrated in Fig. 15 and Fig. 16. It is used on slab-charging cranes, screw-downs, mill tables, etc., where it is subject to very rough handling, intense heat, severe over-loads, dirty surroundings, and other unfavorable conditions. All portions of these motors, such as the frame, shaft and bearings, are built unusually heavy to withstand safely the shocks to which they are subject.

31 There are thousands of other motor applications, which the writer will not attempt to describe. In the very complete paper by Mr. DeLeeuw, The Economy of the Electric Drive in the Machine Shop, are enumerated some of the important points in regard to applying motors to machine tools. The writer has, therefore, endeavored to confine himself to a consideration of these mechanical features not already covered.

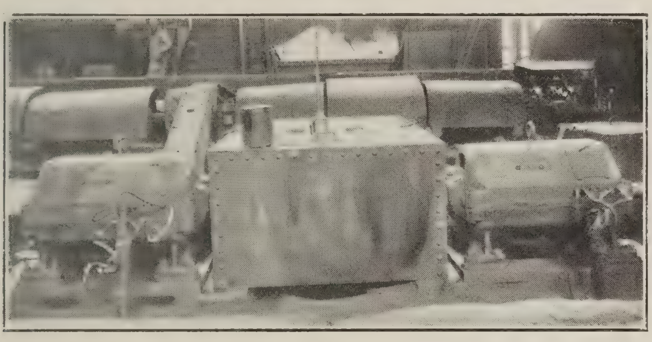

Fig. 16 Two 50-h.p. D.C. Motors

32 At the Schenectady plant alone we have running some 8500 machine tools, of which 8150 are individually motor-driven. Group-driving has been adopted for some sensitive drills, speed lathes, and other small miscellaneous machines. Of the above tools there are 48 portable machines, consisting of slotters, milling and drilling machines, radial drills, etc. These machines are operated on 32,675 sq. ft. of iron floor plate, in addition to which there are 7000 sq. ft. of iron rails, cemented into the floor, for erecting purposes.

33 Had we continued our extensions since 1899, using lineshafts, countershafts and belting, we would have approximately 34,570 ft. of lineshaft, or about $6\frac{1}{2}$ miles, and about $4\frac{3}{4}$ miles of countershaft which would require about 21,225 hangers and bearings. Allowing two belts to each machine, with an average length of 25 ft. per belt, we would have for the 8500 machines a total of 425,000 ft. of belting, equal to about $80\frac{1}{2}$ miles.

DISCUSSION

AN ELECTRIC GAS METER

By Prof. Carl C. Thomas, Published in The Journal for December 1909

ABSTRACT OF PAPER

This paper describes a meter for measuring the rate of flow of gas or air which can be adapted for use as a steam meter or as a steam calorimeter, taking the quality of all the steam passing through a pipe instead of that of a sample of steam. The operation of the gas meter depends upon the principle of adding electrically a known quantity of heat to the gas and determining the rate of flow by the rise in temperature of the gas (about 5 deg. Fahr.) between inlet and outlet. The meter consists of an electric heater formed of suitable resistance-material disposed across the gas passage so as to impart heat at a uniform rate to the gas. The resulting rise of temperature is measured and autographically recorded by means of two electrical-resistance thermometers, one on each side of the heater. These consist of resistance-wire wound upon metal tubes so placed that all the gas passing through the meter comes in close proximity to the thermometers. The adoption of this principle of operation permits the construction of a very accurate and sensitive autographic meter of large capacity containing no moving parts in the gas passage; it is independent of fluctuations in pressure and temperature of the gas and capable of measuring gas or air at either high or low pressures or temperatures. The electrical energy required is about 1 kw. per 50,000 cu. ft. hourly capacity, at the pressures ordinarily used in gas mains.

DISCUSSION

Prof. L. S. Marks. The meter described by Professor Thomas should prove a valuable addition to the instruments used in gas engine and other testing. The possibilities of error in the indications of such an instrument must be fully examined.

2 This meter is fundamentally an instrument for determining the weight of gas or vapor flowing through it and is made to record volumes. It is obvious that these volumes cannot be those actually flowing but must be the volumes reduced to some standard conditions of temperature and pressure. The author has not mentioned this matter in his paper, but it is of considerable importance. A variation of 5 deg. fahr. in temperature, or of 0.3 lb. in pressure, under ordinary

atmospheric conditions, would result in an error of 1 per cent in the indications of the instrument if it were assumed to record actual volumes flowing. The calibration of the instrument by passing through it a known volume of a gas at known pressure and temperature, can easily be reduced to a calibration under standard pressure and temperature conditions.

3 In Par. 16 the author refers to the effect of water vapor carried in with the gas. He states that, in consequence of the small rise of temperature, the water vapor does not experience a change of state, and that, consequently, the latent heat of vaporization does not enter into consideration. It is obvious that he is considering here the case of a gas which not only is saturated with water vapor, but also is bringing with it minute particles of water in suspension. Under these conditions—and they are conditions which may easily obtain with blast-furnace gas which has just passed through the washers—the indications of the instrument will be rendered completely useless.

4 If the gas should enter at a temperature of 70 deg. fahr. it would contain 0.001148 lb. of water vapor. After passing through the meter with a rise of temperature of 5 deg. the same weight of gas could contain 0.001198 lb. of vapor; that is, there would occur a vaporization of 0.00005 lb. of moisture for every cubic foot of gas passing through the meter. The latent heat of vaporization at these temperatures is about 1050 B.t.u., or, 0.0525 B.t.u. will be used in converting the water into vapor. As the total heat required for raising one cubic foot of the gas 5 deg. fahr. is only about 0.1 B.t.u., we have here, obviously, the possibility of an error of the magnitude of 20 or 25 per cent in the indications of the instrument in the case suggested by the author where the gas is supersaturated with vapor.

5 The accuracy of the instrument depends primarily on the accuracy with which the volumetric specific heat of the gas can be determined, and upon the constancy of this quantity while the meter is in operation. For the correct determination of the volumetric specific heat it is necessary to know the volumetric composition of the gas and the volumetric specific heat of each of the constituents. The author has stated that the volumetric specific heat of each kind of gas is very nearly constant and the calibration of the instrument is based upon that assumption; that is, it is proposed to calibrate the instrument with, for example, producer gas, and then to use that calibration when the instrument is used at other times with producer gas. It will be interesting to examine how nearly correct this assumption

is. In the December number of The Journal of the Society there are given four analyses of producer gas, three of them by Mr. Bibbins, and one by Messrs. Garland and Kratz. I have worked out the volumetric specific heats of these gases, using the physical constants given by the author, and I have also taken at random two analyses from tests which I have made on a large anthracite gas producer. The results of the calculations are as follows:

6 For the two lignites in Mr. Bibbins' paper, the values of the specific heats are 0.01920 and 0.01899, which agree very closely with the average stated by the author. For the bituminous coal in Mr. Bibbins' paper, the value is 0.01899, and for the bituminous coal in the paper of Messrs. Garland and Kratz, the value is 0.0186. My own tests with anthracite give values 0.01826 and 0.01848, respectively.

7 It is quite evident from these figures that there is considerable variation, which may be as great as 5 per cent in the volumetric specific heat of producer gas. It may possibly be, as these figures seem to indicate, that the specific heat can be stated with greater accuracy if the type of coal is also specified, since there seems to be a relation between the volatile contents of the coal and the specific heat of the producer gas; but this point has not been sufficiently investigated to permit of any definite conclusions.

8 I have attempted also to see whether the value given for illuminating gas is constant. Only one illuminating gas was considered—that in Cambridge, Mass.—the analysis having been made by the chemist of the gas company. The specific heat calculated from this analysis is 0.02278. The specific heat calculated by the author is 0.02111. The value which he states as being practically constant for illuminating gas is 0.020. There is a variation of over 10 per cent between these values, so it would seem that it is not practicable to calibrate this instrument with illuminating gas at one place and assume it to be accurate when used with illuminating gas at some other place.

9 Moreover it must be recognized that such analysis as that given by the author for illuminating gas is only approximate; the heavy hydrocarbons are never fully analyzed and some kind of guess must be made as to their composition and specific heats. It cannot even be accepted as true that a calibration made with any particular illuminating gas will hold at some later date for gas from the same source. I have found variation in the composition of the Cambridge gas which would certainly cause a variation of two or three per cent in its specific heat.

10 It appears to me then, that this instrument cannot be accepted for accurate measurement unless analyses are being made of the gas that is going through the meter. In scientific testing, such analyses will naturally be undertaken and consequently the instrument should be extremely valuable in such cases. I would like to know what experience the author has had with this instrument in the measurement of volumes when the flow is variable as, for instance, when gas is flowing through a single-acting, four-cycle gas engine. In this case the flow will occur approximately for only one-fourth of the whole time of the test. The author's contention that the indication of the instrument would be accurate under these circumstances seems reasonable, but it would be valuable to know whether, and to what extent, his statement has been verified by actual investigation.

Prof. W. D. Ennis. I do not quite follow Professor Thomas' explanation that the proper correction has been made for fluctuations in the pressure of the gas. A change of, say, five per cent in the pressure, measured above the zero of pressure, would correspond roughly with a change of five per cent in the absolute temperature, without any addition whatever of heat. A change of five per cent in absolute temperature would mean a very large change in Fahrenheit temperature.

2 A more important point is suggested by the statement in Par. 4: "These thermometers consist of wire wound upon vertical tubes so disposed as to come in contact with all the gas passing through the meter, thereby indicating the average temperature over the cross section of the gas passage." If that is what the thermometers do, I question whether they indicate the average temperature of the gas, because more gas is passing at a point in the middle of the pipe than at points near the circumference. Do the thermometers indicate the average temperature of the whole weight of gas, which is the temperature that we must have in order to calculate the weight of gas flowing?

Edwin D. Dreyfus. Certain fuel gases—particularly blast-furnace, coke-oven and producer gas—carry with them a considerable quantity of finely divided solid matter, which in turn forms deposits in the piping or in any piece of apparatus through which the gas passes.

2 From their construction, it would seem that the grids in the meter would favor the formation of deposits of this sort, and I would like to ask whether Professor Thomas has made any trials to deter-

mine what effect, if any, such deposits have on the accuracy and general reliability of the instrument.

3 In cases where the gas is carried long distances through overhead mains—as in many blast-furnace plants—the temperature of the gas will be largely influenced by the temperature of the atmosphere, as between the summer and winter months the gas temperatures might easily vary as much as 50 deg., and the variation in temperature would have a decided effect on the moisture content. It seems probable that the moisture content of the gas is the most disturbing factor affecting the accuracy of the instrument. If this be so, then it is desirable that the actual significance of this factor should be determined by trials made over as wide a range of conditions as we may reasonably expect to meet in ordinary everyday practice.

A. R. Dodge. I would like to ask Professor Thomas if he has made calculations in regard to the amount of power necessary to operate this meter when used as a steam meter. The specific heat of steam being greater than that of gas and air, the amount of power required is considerable. For instance, Thomas meters on the large turbines of the New York Edison Co. would require about 545 kw. at normal load, quite a percentage of the total output of the turbine.

The Author. Bearing upon the questions asked in the discussion, I would say, that in addition to the description of the meter given in the paper, I have given in Fig. 10 completed curves[1] showing the results obtained in calibrating the meter with both illuminating gas and air, reduced to standard conditions of 29.9 in. mercury and 62 deg. fahr. These curves show the method of using the meter for measuring directly standard cubic feet of gas or air at some convenient assumed conditions of pressure and temperature. In Par. 14, line 7 should read ". . . . at mean atmospheric pressure and 60 deg. fahr. . . ." The calculations in the table are for conditions of 760 mm. and 0 deg. cent. The results of measurement by the method described in the paper may be considered as given either in standard cubic feet, or in weight of gas passing the meter.

2 These meters are essentially applicable to the measurement of a dry gas or steam, that is, a gas or steam which is either saturated or superheated. Our experience with the gas meters has thus far been with illuminating gas and with air, and these are exceedingly easy of measurement. The gas or air we are measuring is saturated,

[1] Addition to paper, published in The Journal for January, 1910.

carrying its full quota of water vapor. The smallest quantity of heat introduced causes an immediate rise in temperature of the gas. If the gas carried a spray or mist of water, the measurement would be in error to a certain extent, because of the difference in specific heat between the water vapor and the gas. The extent of the error would depend upon the percentage of water vapor present.

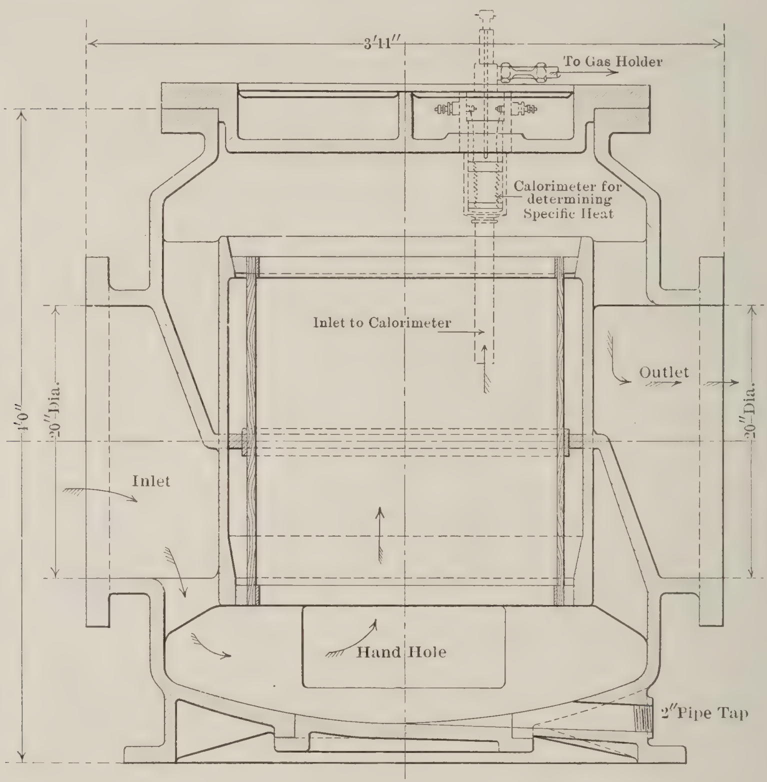

Fig. 1 Gas Meter arranged with Calorimeter for Determining Specific Heat

3 For gas or air under the conditions existing during the tests, of approximately 60 deg. fahr. and 29.8 in. mercury, and 6 in. water pressure, the correction for water vapor introduces a change in the results of less than ½ of 1 per cent, and has therefore been omitted. For other pressures and temperatures the correction for water vapor can be easily made by reference to the charts commonly used in gas works.

4 An interesting confirmation of the statement in Par. 16 is that the most minute addition of electrical energy to the gas or air causes an immediate rise of temperature.

5 Regarding variation of specific heat, the meter prover shown in Fig. 1, herewith, has been developed. It consists of a small electric heater which is placed in the outlet of the meter and discharges into a portable gas-holder such as is used for proving large meters. By this means a small known quantity of the gas is heated, and the specific heat actually determined by direct measurement. This determination can be made as often as desired until the variation of specific heat and satisfactory average values have been determined. So far it appears that the specific heat in a given installation is practically constant from day to day and from one time of day to another. The fact that it is possible thus to determine the specific heat experimentally affords a most valuable check upon the specific heat determined by calculation from chemical analysis, since the methods used in the latter are at best largely approximations.

6 As to dust and impurities collecting on the heater: A meter now in operation for some months has been used for measuring in the neighborhood of 100,000 cu. ft. per hr. of illuminating gas. The heater has been taken out once, and in handling it a small amount of grease was found on the heater material. Otherwise the interior of the meter was clean. In handling very impure gas it will of course be necessary to clean out the meter occasionally, simply in order to provide sufficient area for the passage of the required amount of gas. All the heat generated in the heater necessarily goes into the gas. The operation of heating and measuring difference of temperature is all accomplished in a very short length of travel of the gas. This perhaps answers the question regarding the heat-insulating effect of deposits which may be formed on the heater. The rise of temperature of the material of the heater is only 15 or 20 deg. fahr. This temperature rise might be effected by a considerable deposit on the heater, but the heat generated must necessarily be liberated from the heater and given up to the gas, resulting in no error in gas measurement.

7 As to variable flow, the best evidence is presented by the curves and calculations on the chart. The entire regularity of operation, during experiments conducted under circumstances very favorable to accuracy of observation seem to show that no error is introduced by non-uniformity of flow. If such a cause of error existed, it seems probable that it would have been found during experiments such as have been made with this meter, covering the wide range of from

6000 cu. ft. to about 127,500 cu. ft. per hr. The meter is now being built so that the gas passes in a vertical direction through the heater and thermometers, and this would seem to favor regularity of distribution over the cross section of the passage. The change from horizontal to vertical position was however dictated by convenience of attachment and in order to obtain accessibility, although it seems favorable to the above-mentioned consideration.

8 During the air tests extensive fluctuations of pressure took place, due to the pulsations of the blower supplying the air. These were so great at times as to cause the water to be thrown completely out of the pressure gages, but the results obtained remained entirely regular, as shown on the chart. A small meter has been used on a single-acting three-cylinder four-cycle gas engine delivering from 30 h.p. to 60 h.p. The meter was constructed of sheet iron, and although the pressure fluctuations were such that the sides of the heater "panted" continuously, the measurement of gas was accurately accomplished.

9 Answering Mr. Dodge's question regarding the amount of energy required to measure steam with these meters, we are using 5 deg. fahr. temperature difference, which can be measured to an accuracy of 1 per cent and the energy required is 1 kw. per 1000 lb. of steam per hr. Taking a water rate of 12 lb. per h.p.-hr., 1 kw. would measure the steam used for about 80 h.p.

10 Stated generally this meter seems to be particularly suitable for the measurement of dry saturated or superheated gas, air or steam. The substance to be measured should be dry, but it may be of any pressure and temperature which the materials of construction will stand, and the measurement is independent of fluctuations of pressure and temperature. The recording mechanism can be placed in any convenient position, as, for instance, in an office, instead of near the meter, and the graphical record is thus continually observable. It is not necessary that a graphical record should be taken. An ordinary integrating wattmeter showing the amount of energy it has required, to maintain the constant temperature difference of 5 deg. between inlet and outlet of the meter, suffices as a record of rate of flow, though the variation is best shown by an autographic record

GOVERNING ROLLING MILL ENGINES

By W. P. Caine, Published in The Journal for Mid-November

ABSTRACT OF PAPER

The paper describes first the types of rolling mills and the engines driving them; analyzes the distribution of power, the design and the peration of the engines, calling attention to causes of low steam economy, high repair charges and the danger of broken flywheels; describes and gives indicator cards and speed curves of a Corliss engine driving a three-high mill under two different conditions of governing, (*a*) under the widest range of adjustment of cut-off, (*b*) under a limited range, increasing the economy and making the engine run much more smoothly and safely. It also gives the reasons for the different results shown. A table is given showing the power required for rolling in the mill and the momentary source of the energy, whether from the cylinder or flywheel. A diagram shows this graphically. A description is also given of the tachometer used to take the speed curves.

DISCUSSION

Henry C. Ord. The conservation of energy as applied to rolling mills has received very little attention until during the past five or six years. The power required to roll a given piece was not known until the continuous indicator and recording tachometer were applied. The cards from these instruments furnished records from which the conditions for any stage of the operation could be calculated, giving complete information as to the variation in power and speed for different conditions and classes of work.

2 Rolling mill engineers have several reasons for preferring the two-high mill. As Mr. Caine says: "The engine uses steam only when the piece is on the mill." As there is considerable time between pieces in some classes of work, this is an important item. Should a piece not enter properly and stick in the rolls, thus stalling the engine, it is easy to reverse and back out the piece. This condition with a three-high mill would cause considerable trouble and delay. This is the reason why some of the modern three-high electric-motor-driven rolls are fitted with an emergency reversing de-

vice. The reversing feature can also be used as a quick safety-stop in case of accident.

3 The two-high mills are not so complicated as the three-high mills, and they have less rolls and no reversing mechanism for raising and lowering the table. In considering the two systems, this is an item of power that should be charged to the three-high mill. However, power is not the only consideration; it is usually a question of the maximum tonnage in minimum time with the least amount of power.

4 The 25,000 h.p. engine Mr. Caine refers to is a 42 in. and 70 in. by 54 in. twin tandem horizontal compound-condensing blooming-mill engine, designed by the writer about four years ago, and built by the Allis-Chalmers Company for a blooming mill requiring an average of about 6000 h.p., which is also about the economical load for the engine. It was designed for a maximum of 25,000 h.p. under the following conditions: steam pressure, 150 lb. gage; cut-off, $\frac{3}{4}$ stroke; vacuum 25 in. referred to 30 in. barometer; r.p.m. 200. This machine has been described as "the world's most powerful engine." I believe the piston speed, 1800 ft. per min., is the world's record.

5 If the engine Mr. Caine has experimented on was tested under the same conditions as regards pressure and work, with and without the function of the adjusting screw, we would expect different results than those shown. The controlling device has no control over the engine before the load is increased, until the speed falls to that fixed by the adjusting screw. At this speed and power the engine will be doing the maximum work allowed by the adjusting screw; consequently this control can be applied only to engines that have a longer-range cut-off than is required for the greatest loads they have to carry. After the above conditions are studied, it will be evident that to prevent the engine's being stalled before reaching the latest cut-off for which it was designed, we would have to dispense with the services of the adjusting screw.

6 From a study of the speed curves in Fig. 2, assuming that the height of the governor varies approximately as the speed of the engine, it will be seen that had the adjusting screw been applied and adjusted for the maximum load or minimum speed, it would be in momentary control during the second pass, and from the speed curves given in Fig. 3, it is seen that it was in action for about the same length of time during the third pass. As it would take considerable time for sufficient change in the energy of the flywheel to produce the results claimed, we believe there are other reasons for the improved condi-

tions shown. When the adjusting screw is in control, the engine will slow down much more quickly than without it, and the engine would be stalled by a lighter load; it would also take more time to do a given amount of work.

7 As engineers prefer to have engines with some power in reserve to take care of the abnormal load, I believe they would hesitate before using any method of control that eliminates the reserve power of the motor to which it is applied.

8 From a study of the Indiana Steel Company's plant at Gary, Ind., it is evident that conservation of energy as applied to steel plants has received considerable study. The rolls are driven by motors, current being supplied by gas-engine generators.

James Tribe. In Par. 5, Mr. Caine refers to a certain engine capable of developing 25,000 h.p. while the average load does not exceed one-seventh of its maximum capacity. I do not know what engine he refers to, but a blooming mill engine of unusually large dimensions and answering somewhat to the description given, was built by the Allis-Chalmers Company and installed less than two years ago at the Carnegie Steel Company's South Sharon plant. This was a reversing engine for rolling 28 in. by 28 in. ingots on a two-high mill. The maximum power, or rather, the maximum possibility, of this engine, was likewise 25,000 h.p., which was also far in excess of its average load, but it is doubtful if there is in existence a more efficient reversing blooming-mill steam engine equipment.

2 In Par. 6, Mr. Caine asserts that in a three-high mill driven continuously in one direction, the energy stored in the flywheel would make it possible to do the same work with considerably less than one-half the power. There should therefore be some explanation to justify the installation of so large an engine, at so recent a date, and having so large a percentage of surplus capacity. There are two reasons for this: first, because of the stalling action at the moment the rolls bite the ingot; secondly, because of the probable increase of speed as the ingot is released.

3 The reversing engine, for well-known reasons, has no flywheel, consequently the momentum of the rotating parts is comparatively nothing. Therefore the stalling action at the instant of biting the ingot, due to the tremendous impact, which is followed immediately by an abnormally high tangential resistance at the rolls surface, creates a demand for an exceedingly powerful engine. It is just at this moment that surplus power, or reserve energy, is of the most

vital necessity in order to save time and heat which would otherwise be wasted while waiting for the engine to recover itself. At this critical moment the term "horsepower" does not explain the measure of effort necessary for overcoming this resistance; for as a less powerful engine would be almost, if not quite, brought to rest, two of the power elements, namely, time and space, are for the time being practically eliminated, and the engine reduced to a simple "force" acting on the crank pin. Hence, it becomes a question of a turning moment sufficient here to overcome the resistance, and of regaining normal speed in the shortest possible time: for loss of time means not only delay (which is very serious), but loss of heat, and loss of heat means additional power necessary.

4 In the second place, the engine must be so constructed as to be capable of permitting 25 per cent increase of speed above normal with perfect safety, for the reason that at the instant the ingot leaves the rolls, the slightest delay on the part of the operator in shutting off steam, all resistance except friction having been suddenly removed, results in an increase of speed and the safe limit is quickly reached. These two extreme conditions, full steam and abnormal speed, never occur at the same instant, in actual operation, but the engine must be capable of meeting them, and therefore such an engine may be said to be capable of several times its normal capacity.

5 So far as the gripping and the releasing of the ingot are concerned the effect is the same whether a reversing or a continuously running engine is employed; for the energy of a flywheel may to some extent prevent the stalling action, just as this is accomplished by the surplus capacity of the larger engine. But flywheel energy cannot be spent without a proportionate reduction in speed, and with loss of velocity more time must be taken to regain it than would be the case where the force of the steam is applied entirely in the mill. Part of the steam energy would be spent in restoring the wheel energy and consequently, more time would be consumed in the pass than is the case in a sufficiently powerful engine without a flywheel. This loss of time and heat partly offsets the apparent gain in economy of the smaller engine. But the more serious loss would be experienced in a three-high mill, in both time and heat, as well as the additional power required for raising and lowering the ingot to the two different levels for each succeeding pass. Considering the shortness of the passes in blooming-mill work, this delay would be a very serious loss.

6 It therefore seems to me that but little, if any, substantial advantage can be gained in heavy blooming-mill work by the three-

high mill so long as it is steam-driven. It also seems to me that the only hope of any improvement in economy over present practice will be in the use of the present two-high reversing mill, but driven electrically. In such an equipment, we would have the necessary power to avoid delay on gripping the ingot, the means for instantly throwing off the power at the release of the ingot, and also the continuously running steam engine with a sufficiently heavy flywheel at the generator.

E. W. Yearsley.[1] The value of the flywheel as a means to obtain constant load with intermittent work is well illustrated by Mr. Caine's experiments. This arrangement has been considerably developed in conjunction with electrically driven rolling mills. Where considerable speed variation is allowable, and there is a suitable ratio of pause to operation time, the flywheel may be applied to many drives with economy.

2 Economical considerations are at present of great importance in the steel industry. Engines used for driving rolling mills are usually excessive steam consumers. There is no doubt that their performance in this respect can be greatly improved, especially for continuously running mills. In my opinion the electric motor will be found more reliable and satisfactory for this work, and it will be desirable to confine the refinements necessary for great economy of prime movers to an electric generating station.

3 Mr. Caine's method of regulating the governor is somewhat analogous to that used for controlling the rate of application and the limit of electric current to a main roll motor, in order to obtain the similar results of more uniform load, less rapid speed variation, and protection of the driver. The tests show conclusively the improvement in steam consumption and performance resulting.

4 As the paper points out, the problem is considerably complicated by variation in the number of pieces passing simultaneously, also by variation of the interval between passes and its relation to the time of the pass, and in the temperature and composition of the material. A speed variation of from 12 to 20 per cent transferring from 23 to 36 per cent of the kinetic energy of the flywheel, has been found desirable. With a given torque, time of load, and interval, this speed change fixes the weight of wheel required. Data of power performance of rolling-mill drives are rapidly accumulating. This paper is an interesting addition to such information.

[1]Electrical Engineer, Midvale Steel Company, Philadelphia, Pa.

The Author. Mr. Ord seems to have the impression that there was a great difference in the work done in the two examples given. As a matter of fact, the area of the piece was the same in each case, on entering the first pass, and therefore, the total work for the four passes would be as their relative weights, 2680 lb. for Case A and 2550 for Case B; B having a slight advantage in weight, and A an advantage of 5 lb. in steam pressure, so that the work was practically the same in each case.

2 The valve setting was not altered between tests. The difference in the behavior of the engine was due to the adjusting screw alone; and now, three years after these tests were made, this screw is still in service. This method of engine control does not eliminate the reserve power; it does cut it down to a point where judgment says there is still sufficient reserve to answer all requirements.

3 Mr. Tribe asks the reason for building reversing engines with such a large surplus of power. Such engines are usually driving blooming mills, where it is no uncommon practice to roll about one-half of the total number of passes, from bloom to finished product, in one stand of rolls, the remainder being taken care of from three, four or more stands of rolls, so that the blooming mill must handle these passes in very rapid succession in order to get the tonnage. The engineer handles the throttle and reverse levers, and the roller, the screw-down and the table rolls. The screw-down adjusts the distance between the rolls; consequently it fixes the amount of reduction on the bloom and the load on the engine is proportional to the reduction.

4 The screw-down has no fixed limits for each pass, therefore it will be set in a very short period, according to the judgment, or lack of judgment, of the roller. The writer has timed these operations with a stop watch and found that quite often the adjustment was made in less than two seconds; that is, the time from the end of one pass to the beginning of the next. It is quite likely that the screw-down does not get located where the operator intended; if the reduction is less, the roller will make some other passes heavier because he does not wish to add two additional passes. From the calculated results, from continuous indication cards on an engine of this type, on a single bloom one pass was noted where no reduction was made, while another pass required nearly three times the average power. From this sort of operating conditions, coupled with the desire to get an engine that will not stall under any circumstances, it becomes very evident why there is a great surplus of power. This also calls atten-

tion to one of the features in favor of the three-high mills, namely, that the roll designer can distribute the work approximately equally on every pass, with the proper data at hand.

5 The fact that the reversing engine is man-governed is brought out. This practically places the speed limit at the rate at which it would run with a wide open throttle and nothing in the mill; which would far exceed 25 per cent of the normal. Speed curve A shows that our engines run at about 16 per cent above normal, and with curve B at but 10 per cent above.

6 Mr. Yearsley suggests that the principle involved might be applied to other than mill engines. The writer can cite an instance where this was done. Our company has two-crank flywheel hydraulic pumps which are started and stopped by an accumulator. When the accumulator would drop, the governing throttle valves would open wide and the pumps would run up to the speed determined by the fly-ball governors (50 r.p.m.), and when the accumulator reached the top limit it would shut off the steam, stopping the pumps very abruptly. This continual starting and stopping caused considerable trouble in keeping up the various adjustments, and pins ran hot at times. Upon my suggestion the engineer in charge adjusted the governing throttle valves so that they could be only partially opened, and as a result the maximum speed is just a little above the average, the pumps running almost continually at about 20 r.p.m., the trouble with hot pins is no longer experienced, the rod adjustments last several times as long, and it is my belief that the water valves must give less trouble.

EFFICIENCY TESTS OF STEAM NOZZLES

By Prof. F. H. Sibley and T. S. Kemble, Published in The Journal for Mid-November

ABSTRACT OF PAPER

The object of the tests was to determine the efficiency of various shaped nozzles with steam flowing from a given initial pressure to a known vacuum; also to determine the effect on the efficiency of changing the angle of divergence.

Two methods were tried out for finding this efficiency: (*a*) by first finding the pressure in the nozzle by means of a search tube placed axially in the nozzle; (*b*) by finding the reaction of the nozzle. This was done by suspending the nozzle in an air-tight box at the end of a flexible steel tube. The deflection of the tube caused by the reaction of the nozzle was measured by a calibrated spring. Friction was eliminated. Preliminary work was done to calibrate the springs, to determine the volume of flow of the various nozzles and to determine the pressure in the nozzle and the surrounding medium.

The results of the tests indicate: (*a*) that the reaction is affected by a difference in pressure between the muzzle of the nozzle and the medium surrounding the nozzle; (*b*) that the efficiencies of the various nozzles were determined within a probable error of 2 per cent: (*c*) that the efficiency is affected more by the smoothness of finish on the inside of the nozzle than by the exact contour of the nozzle.

DISCUSSION.

Prof. J. A. Moyer. The methods used in these tests are obviously much more accurate than the impact plate devices used by Lewicki in his experiments with De Laval nozzles and by others who have conducted similar investigations more recently.

2 The high efficiencies obtained may be surprising to some who have not followed the latest developments in the designing of steam nozzles. Results of this investigation confirm in general the results given by Steinmetz[1] and by the writer showing that the efficiency of a well-designed nozzle for relatively large, as well as for small, limits of pressure will be above 97 per cent.

3 However, in one respect the investigation is not as complete as it was hoped it would be. There are not enough data to determine the effect on the efficiency of varying the length of a nozzle: that is,

[1] The Journal, Am. Soc. M. E., May 1908, p. 628.

nozzles of different lengths, but with the same taper or angle of divergence, should be compared. However, the statement is made in the last paragraph of the paper that there is no appreciable difference in the efficiencies of nozzles 10, 11 and 12, which, however, do not have the same taper, but have the same *areas* at the throat and at the mouth. It is probable that all of these nozzles were longer than they should be to obtain the highest efficiencies. More data are needed about the best length of the nozzle for a given expansion. Lewicki's experiments cover the two extremes: nozzles which are obviously too short, and those which resemble in proportions the ones used in this investigation.

4 The error due to moisture in the steam could not readily be determined, and while it is probably not large, yet this uncertainty might have been avoided by using superheated steam. The reaction in a nozzle due to the flow of superheated steam is apparently constant for a varying amount of superheat. This can be shown by the usual thermodynamic equations for flow and velocity—which determine the impulse force of a jet— and by the experiments of Lewicki[1] on the flow of superheated steam through De Laval nozzles. It should be observed however, that when these tests were started, Knoblauch and Jakob had not yet published the values which we are now using for the specific heat of superheated steam, and for this reason alone it was desirable to avoid the use of superheated steam.

5 It has not been mentioned by the authors of this paper that their method can be used to calculate the *apparent* efficiency of any nozzle for any initial and final pressures. By measuring the areas of a nozzle at the throat and at the mouth or "muzzle," the expansion ratio in a nozzle is determined, and by means of empirical equations, due to Zeuner and others,[2] the ratio of the corresponding initial and final pressures giving the highest efficiency, can be obtained. This ratio of pressures would correspond to the condition in these tests where the terminal and box pressures are the same.

6 If the ratio of the initial to the final pressure has been determined, either of these pressures can be readily calculated if the other is known. For example, if by measurement of the mouth and the throat areas, the expansion ratio of the nozzle is found to be, say, 3, then the ratio of the initial to the final pressure must be nearly 13.3 for the maximum efficiency of the jet discharged from it. For

[1] Mitteilungen über Forschungsarbeiten, Heft 12, Zalentafel 9 (*c*). Verein deutscher Ingenieure, 1904.

[2] J. A. Moyer, The Steam Turbine, p. 40–41.

this nozzle, therefore, with an initial pressure of 200 lb. absolute, the final pressure should be 15 lb. absolute. From the equations given in Par. 4 of the paper, the theoretical reaction can be readily calculated from the available energy corresponding to the pressure limits. The change in reaction due to final pressures different from those for which a nozzle is designed is, then, according to the method presented here, the product of the area of the mouth of the nozzle, times the difference between the correct final pressure for the nozzle—in this case 15 lb. absolute—and the pressure in the box, or in practice the pressure inside the casing of a stage of a turbine. Since reaction and velocity are directly proportional—*with constant flow*—the apparent velocity of the jet will be increased or decreased in the same proportion as the reaction is increased or decreased.

7 In actual practice, however, this does not occur. It has been observed that if a nozzle is used which does not expand the steam sufficiently, there is not nearly so much loss in the velocity of the jet as when the nozzle is too wide at the mouth and "over-expands" the steam. In other words, it has been found that a nozzle which is about 25 per cent too large in area at the mouth, will give to the jet only 90 per cent of the theoretical velocity, while one which is too small by the same percentage will give within 2 or 3 per cent of the maximum efficiency obtainable with the pressures best suited. All this involves something which is not taken into consideration in these reaction experiments; and for that reason, the results obtained by this method with varying back-pressures may possibly be misleading.

Prof. C. C. Thomas. I have for years been interested in this line of investigation, and am glad to see this contribution. In Par. 26, the corrections which the authors make to the observed reactions seem to me to be somewhat open to question. Aside from this fact, I cannot quite see the theory upon which the corrections are based; the fact that the pressures vary considerably in all but perfect nozzles, from the center to the walls, and that very considerable irregularities of flow are found in nozzles, makes me doubt the necessity for making these corrections to the observed reactions.

Strickland L. Kneass. The tests appear to cover ordinary straight-tapered nozzles, as follows: 1 in 6, 1 in 5.77, 1 in 5, 1 in 4 and 1 in 3. In several cases the net areas vary slightly from these ratios owing to the displacement of the cylindrical search tube. but

the ratio of the throat area to the outlet area is practically the same for all nozzles, so that the results relate chiefly to the effect of the length of the tubes and the friction upon the walls.

2 From Table 5 it would appear that nozzle No. 13, which has a taper of approximately 1 in 4, gives a much higher efficiency between 100 lb. and 145 lb. absolute, than nozzles Nos. 9, 11 and 14, with tapers ranging from 1 in $3\frac{1}{2}$ to 1 in 6. As far as the knowledge of the writer extends, there is no logical reason for this result, and he would attribute the higher percentages to greater precision in the experiments rather than to any inherent efficiency in the 1-in-4 nozzle.

3 The correct contour of a nozzle for the discharge of elastic fluid is still a moot question. After an extended series of experiments between the years 1888 and 1891 with steam nozzles of various tapers, the writer offered the suggestion that the section should be in the form of a reversed curve, somewhat as shown in Fig. 1 here-

Fig. 1 Suggested Form of Nozzle

with. This curve was based on the theory that the acceleration should be constant during the passage of the steam through the nozzle, and that the areas at the several sections should be unit distances apart. These sections were calculated with due allowance for the change in the specific volume of the steam during expansion. The results obtained seemed to confirm this theory and were compared with the discharge from straight-tapered nozzles in a paper read before the Engineers' Club of Philadelphia in 1891. The writer's opinion was further corroborated by F. Hodgkinson before the Engineers' Society of Western Pennsylvania in 1900. In view, therefore, of published experiments upon nozzles of special contour for which advantageous results were claimed, it is surprising that the authors of this paper did not increase its value by widening the scope of their experiments, instead of confining their tests to the oldest and possibly less efficient form of tube.

4 Referring again to the experiments of the writer, his conclusions covered the general theorem that there was little difference

in the efficiency of the straight-tapered nozzle, so long as the terminal pressure of the steam within the tube was the same as that of the medium into which it flowed, and, further, that the terminal velocity would be the same under this given condition whether the taper were 1 in 6, 1 in 5, or 1 in 3. This opinion seems to be sustained by the authors, although the results are not satisfyingly definite, because different terminal pressures were used with each initial pressure and the table does not contain the terminal pressures within the nozzle, so that the comparison cannot be made with the pressure of the final medium.

5 It is desired that this point be emphasized, for a slight difference between these two pressures has an important effect upon the results. It is thought that a more exact method of determining the relative efficiency would have been to modify the length so that the terminal internal and external pressures would always be the same, for when an attempt is made to introduce minus or plus reaction for correction, doubt is thrown upon the result. This is especially obvious to any one who by careful observation of the flow of steam through and from nozzles of different proportions, has noted the unstable equilibrium of the jet when the terminal pressure of the medium exceeds that within the end of the nozzle. Some of the minor discrepancies may be charged to this item and the writer is somewhat skeptical as to the accuracy of the results obtained in practice when calculated under the theorem given in Par. 26.

6 It would have been interesting if the authors had recorded new data relative to the action of the steam within the nozzle and determined the terminal specific steam volume. The writer maintains that the specific gravity of steam at different sections of the nozzle does not correspond to that calculated by the thermo-dynamic equation, and therefore would be glad to have the authors state if the velocity of the steam, as given in Tables 3 and 5, is equal to the specific volume based upon the adiabatic equation, divided by the cross-sectional area.

7 A test of this kind should give the initial condition of the steam. The authors state that a thermometer was placed in a well at the rear of the nozzle, but there are no figures in the table giving the percentage of moisture. An objection to the construction of the apparatus can be offered in the liability of condensation of steam in the vertical flexible supply pipe. The steam flows downward under pressures varying from 100 lb. (328 deg. fahr.) to 145 lb. (356 deg. fahr.) and is surrounded by steam at a pressure

of 28 in. vacuum (100 deg. fahr.) so that a certain amount is sure to be condensed.

The Authors. It appears to be a generally accepted fact that under-expansion in the nozzle is preferable to over-expansion. Stodola's Theory of Steam Shock and his search-tube experiments point very decidedly in this direction Reaction experiments may even *appear* to indicate that under-expansion in the nozzle is in some cases preferable to using the theoretically correct ratio. This may also be true; but if the theory advanced in Par. 26 is correct, it is impossible to accept the results of any purely reaction experiments as giving a definite answer to this question; and where the pressure in the muzzle of the nozzle is not taken into account, all the results may be in error.

2 Of course, it is possible to calculate the muzzle pressure by theoretical and empirical formulae; but if we are to rely upon theoretical formulae there is no object in conducting tedious and expensive experiments. Moreover, empirical formulae on this subject are at least liable to be based in part upon reaction tests which have not taken into proper account the pressure in the muzzle of the nozzle. Also when the nozzle discharges into a pressure which is considerably greater than the theoretical muzzle pressure, violent fluctuations occur within the nozzle itself, so that the formulae do not apply and the results of reaction tests may become very misleading.

3 Par. 26 has been called in question from both the theoretical and the practical standpoint, so that a more extended consideration may not be out of order.

4 The first statement, "The reaction of any nozzle is equal to the summation of all the components, parallel to its axis, of the pressures within the nozzle and in the chamber from which it leads," can scarcely be questioned.

5 The net accelerating force F (Par. 4) which produces the velocity actually present in the muzzle of the nozzle may be divided into two parts. One part (call F_f) is a summation of components of the forces with which the internal walls react against the pressure of the steam. The second part is a force due to the pressure of the steam in the muzzle, and acts in opposition to the first.

6 Let F_m be this second part, P_m the muzzle pressure, and A the muzzle area. Then

$$F_m = P_m A$$

and

$$F = F_f - F_m = F_f - P_m A$$

Let R be the "true reaction of the nozzle," i. e., the force which is equal and opposite to F. Then

$$R = F = F_f - P_m A \tag{1}$$

7 The apparent reaction (called R_a) is the force with which the nozzle actually pulls or pushes in the direction opposite to the steam flow during the test. The apparent reaction of any nozzle is equal to the summation of the components parallel to its axis, of all the pressures, both internal and external, upon the walls of the nozzle and of the chamber from which it leads.

8 That part which is due to the internal wall pressure is equal to F_f. The external pressure acts, in the direction of flow of the jet, upon an area which is greater than that upon which it acts in the opposite direction, the difference being the area of the muzzle.[1]

9 Let P_e be the external pressure. Then

$$R_a = F_f - P_e A \tag{2}$$

Combining (1) and (2) we have

$$R = R_a = A\ (P_m - P_e) \tag{3}$$

10 The rest of Par. 26 accords with these equations.

11 It is evident from this that any acceleration or retardation of the jet beyond the muzzle (due to the pressure into which it is discharged or to any other cause) cannot affect the true reaction, and that so long as the conditions within the jet are stable so that the muzzle pressure can be properly determined, there is no danger of being misled except by a failure to make the corrections.

12 When the pressure into which the nozzle discharges is considerably greater than the theoretical muzzle pressure, such violent flucuations ensue as to make all corrections impracticable, and the reaction tests under these conditions become worse than useless because they are misleading. The criticism by Professor Thomas is well founded with regard to such cases; but does not apply to the tests reported in this paper for the reason that these were all made under conditions which did not disturb the stability of the jet within the nozzle.

13 The fact that the corrected reactions shown in Fig. 13 and Fig. 14 lie in a horizontal line, i. e., are equal, is a further evidence that the theory upon which they are based is correct, also of the fact that the jet within the nozzle remained in very stable equilibrium,

[1] Gages connected to various points within the box showed that the external pressure did not vary in different parts of the box by as much as 0.01 lb. per sq. in. It must be remembered that the nozzle and the chamber from which it leads are here suspended within the box into which the jet discharges.

and that the creeping in of the external pressure along the internal wall had no practical effect, while the box pressure varied within the limits shown.

14 To show further the form of error involved in the failure to use these corrections, apparent and true reactions have been taken from Fig. 13 and Fig. 14, and the accompanying table computed.

Nozzle	i. p.	Flow Lbs. per sec.	Box t. p.	Reac.	Vel.	B. t. u.$_1$	B. t. u.$_2$ Table	Eff.
11	145	.1536	0.929	18.134*	3796	288.0	317.4	90.75
14	145	.1550	1.632	17.821*	3698	273.2	289.8	94.28
11	145	.1536	1.632	17.01†	3561	253.5	289.9	87.43
14	145	.1550	0.929	18.52†	3843	295.1	317.4	92.96
11	100	.1069	0.638	12.45*	3744	280.1	311.5	89.91
14	100	.1081	1.116	12.295*	3659	267.6	284.7	93.99
11	100	.1069	1.116	11.69†	3517	247.2	284.7	86.81
14	100	.1081	0.638	12.77†	3799	288.4	311.5	92.59

*Apparent and true.
†Apparent.

Data obtained from nozzles No. 11 and No. 14, with the box pressure equal to that in the muzzle of the nozzle, are given in lines 1 and 2. These velocities and efficiencies are the same as those given in Table 5, and require no correction for terminal pressure.

15 For line 3 the apparent reaction is taken for nozzle No. 11 with a box pressure which would be correct for No. 14, and the apparent velocity and efficiency of No. 11 are calculated from that basis.

16 For line 4 the apparent reaction for nozzle No. 14, with a box pressure which would be correct for nozzle No. 11, is similarly used.

17 It was found in the experiments plotted in Fig. 9 and Fig. 10, that the pressure conditions within the nozzle remained stable and practically constant with such variations from the proper box pressure for each nozzle. Also, by applying the corrections called for in Par. 26 it is found that these values reduce to the same values as those obtained in lines 1 and 2, showing that the velocity and efficiencies of the jets as they reached the muzzles were not affected by the changes in box pressure.

18 The acceptance of the uncorrected values would therefore imply an assumption that in nozzle No. 14, with an initial pressure of 145 lb. and a terminal pressure of 0.929 lb., the jet attained a velocity of 3698 ft. per sec. in the nozzle, and that after leaving the nozzle its velocity jumped to 3843 ft. per sec., and that in nozzle No. 11, with

an initial pressure of 145 lb. and a terminal pressure of 1.682 lb., the velocity of the jet after leaving the nozzle dropped from 3796 ft. to 3561 ft. per sec.

19 The efficiencies calculated from the apparent reactions, if accepted in this form, would show that No. 14 is better than No. 11, not only for its own proper terminal pressures, but for the terminal pressures found in the muzzle of No. 11 as well. It may be that such is the case; but there is considerable probability of arriving at erroneous conclusions if it is assumed arbitrarily, without having first been proved by very careful experiments which are not in any manner dependent upon the assumption for their accuracy. There certainly is no basis for making such an assumption from these data as it has no bearing whatever upon the subject.

20 Previous to the time when this series of tests was begun, there had been considerable investigation of nozzles with cone angles up to 12 deg.; but the action of steam in nozzles of greater cone angle had not received the same degree of attention. It was therefore decided to use nozzles with divergence angles of from 9 to 20 deg., it being then thought that this upper limit might be beyond the value for highest efficiency.

21 Another set of nozzles tested contained one with a cone angle of 24 deg. 30 min., which seemed to show an equal efficiency with those of smaller angle. This set was made of babbitt metal, was not perfectly smooth and was somewhat worn with long-continued use, so that the results could not be thoroughly checked.

22 With the steam conditions given and the ratio of muzzle to throat area determined therefrom, the only point left for the designer is the general contour of the nozzle, including the shape of cross section, length and angle or angles of divergence. The two sets of nozzles shown in Fig. 6 and Fig. 7 were designed with this in mind, each set having a common ratio of areas; those of Fig. 7 differing among themselves only in length and consequent angle of divergence, or vice versa, and those of Fig. 6 differing only in elements of general contour, not including length.

23 Professor Moyer's statement that "nozzles of different lengths, but with the same taper or angle of divergence, should be compared," is not understood, unless he means to suggest that the whole field of different steam expansion ratios should have been investigated. This was not permitted because of limitations of time and other circumstances familiar to most investigators. Such an investigation would not serve to determine the proper length for a given steam expan-

sion ratio, because the different nozzles would not be suited to the same steam conditions; but it would give the efficiencies for one angle of divergence with all the pressure ratios to which the various nozzles were adapted.

24 Each set contained one search-tube nozzle for use in determining experimentally the terminal pressure in the muzzle, to be applied in reaction tests on the rest of the nozzles in that set. The efficiencies of these nozzles, No. 9 and No. 13, as calculated by the search-tube method, are shown in Table 5; but they are not worthy of consideration except as an example of the inaccuracies almost certain to be involved in this method. The high efficiency given for nozzle No. 13 is not due to greater precision in the experiments, as Mr. Knease suggests, but rather to the great error in the search-tube method of calculation, caused by a very small error in determining the muzzle pressure. In Table 6 it is pointed out that a "+ error" of only 0.1 lb. per sq. in. in determining the terminal pressure would cause a "– error" of from 5.4 to 14 per cent in the "search tube computed" efficiency of No. 9 and No 13.

25 These "search-tube computed" efficiencies are evidently responsible for Mr. Moyer's statement that efficiencies were here found as high as 97 per cent. Values obtained from reaction tests are lower, and it is upon these that the conclusions stated in Par. 49 are based.

26 No. 9 ("search-tube" nozzle) was made with a small angle of divergence, to be doubly sure that the steam should not leave the walls before reaching the muzzle.

27 Both the length and the ratio of areas in nozzle No. 10 were made to correspond as nearly as possible with those in nozzle No. 9 so that the terminal pressure found in the muzzle of No. 9 might be applied to reaction tests upon the former with the least possible error.

28 No. 11 and No. 12 were made shorter and with a greater cone angle but with the same sectional areas, in order to find out what difference, if any, this would make in efficiency.

29 No. 18 was finished rough for comparison with No. 11, upon which the greater number of tests had been made.

30 No 14 was used to determine the efficiency with a smaller expansion ratio.

31 No. 13 ("search-tube" nozzle) was made to correspond as nearly as possible with No. 14, so that the terminal pressure as determined in the former might be applied in reaction tests with the latter.

32 No. 15 and No. 16 were used to determine the effect of these very considerable variations in contour.

33 Other forms, such as shorter nozzles or those designed for uniform acceleration and upon other theories, may and probably do give just as good efficiency as those herein described. It seems doubtful, however, in view of the uniform results obtained with nozzles of such different contour as those covered by these experiments, whether it would be advantageous to use any form especially difficult to manufacture, unless it be for the purpose of controlling the shape of the jet as it strikes the moving blades of the turbine. This is very important, as it has a great effect upon the efficiency of action in the blades.

34 It is to be regretted, as stated in Par. 19, that we were unable to procure a calorimeter of sufficient accuracy for our purpose, but such great care was taken to maintain uniform conditions in the boiler room, and these conditions gave such repeated indications of the dryness of the steam at the nozzle entrance, that the probable error introduced is not serious.

35 As stated in Par. 14, the steam left the boiler under a pressure not varying more than 2 lb. from 155 lb. gage, and with about 50 deg. fahr., superheat. Steam was throttled to the required initial pressure just before entering the flexible pipe, with the result that the thermometer inserted at the nozzle entrance showed about 4 deg. superheat with 700-lb. flow per hr. and sometimes a trace of superheat with 500 lb. per hr. It is probably fair to assume from this that the steam was dry when used with 145-lb. pressure at the entrance to the nozzle, and that (in view of the greater throttling which tends to offset the increased unit radiation from the pipes) there was always less than 2 per cent of moisture present even with pressures as low as 100 lb. abs.

36 It may be stated in conclusion that a proper method of determining the net effect of under and over-expansion in the nozzle would be as follows:

First: Make a set of nozzles of the same cone angle and finish with throats identical, and with muzzles of different areas.

Second: Determine accurately the proper terminal pressure and the true efficiency of each nozzle, by the method herein described, using a reaction apparatus in which static and moving friction has been eliminated.

Third: Find the push upon a set of turbine blades, using each nozzle discharging into its own proper terminal pressure and into the pressures which are proper for each of the other nozzles of the set.

Fourth: A comparison of the push exerted under these conditions, bearing in mind the "true efficiency" of each jet within the nozzles, will show the net effect of under and over-expansion.

GAS POWER SECTION

THE WORK AND POSSIBILITIES OF THE GAS POWER SECTION[1]

By Fred R. Low, Assoc. Mem. Am. Soc. M. E.

Chairman for 1909

The work of the Section for the second year of its organized existence has been largely formative. The art, to develop and to chronicle the development of which is our avowed purpose,is so recent even in its beginnings, so new in weighty accomplishment, that it is possible for an organization with our facilities to gather and put upon permanent record the main facts of its history and the precedents and data evolved in its development. Our endeavor should be to do this in such a way that another generation will have no regret based upon the failure of ours to make a full and intelligent use of its opportunities.

The Standardization Committee has presented a preliminary report dealing with the significance of various terms used in the art. While nothing that we can do or say will endow any of these terms or quantities with an arbitrary measure or value, your treatment and disposition of the report will go far toward determining the future usage upon which legal and other interpretations will be based.

The Committee on Literature is organizing so as to provide for the systematic reading of current gas power literature, and the filing of index and reference cards with the librarian so that future searchers may find available information grouped for ready reference, and the information itself either in the files or elsewhere in the library. This is not retroactive, however, and some steps should be taken, while present but perishable information is available, to put upon record the history of the art, to make up its complete bibliography, a roster of its personnel and a chronological statement of its achievements; to

[1]Address of the retiring Chairman at the annual meeting.

write the subject of gas power down to date as Professor Dalby has written that of heat transmission.

The Installation and Plant Operations Committees have been occupied in perfecting systems, and the forms submitted for your consideration are parts of such systems, which are thought to coördinate with and to supplement each other and to fulfil all the purposes of the Section. The work of collecting information along these lines will be vigorously prosecuted during the coming year, and any suggestions which will improve the efficiency or practicability of the system will be of especial value at the outset of the work. In addition to these records of regular operation some effort should be made to obtain information as to operating difficulties overcome, and truths as to the behavior of a gas power plant in the hands of the user.

A number of laboratories are available for research work, and the Section is asked to suggest lines of experimentation. A large number of suggestions boil down to 42 determinations which it is desirable to make upon fuels; 8 questions regarding test methods, the determination of which is a matter of debate rather than of experimentation; 20 questions relating to producer practice; 5 to the effects of various factors on engine capacity and efficiency; 11 as to the effect of various elements of design; 6 upon ignition; 4 upon regulation; 3 upon carbureters; 6 upon the heat of the exhaust; 11 upon jacket water; 7 upon operative questions; 16 upon meters and analysis apparatus; 5 on the indicator; and 1 upon fire risk. Not all of these require physical experimentation for their determination, but the Section is fortunate in having received offers of coöperation from several of the foremost interested professional observers and investigators with well-equipped laboratories and skilled assistants at their command, and their thought and work will be invaluable in settling aright these perplexing questions pertaining to a new industry.

The results of the Meetings Committee's work are before you in the papers which have been presented at this and the other meetings of the year

The membership of the Society has increased during the year from 247 to 378, a gain of over 50 per cent.

The development of the past year has been steady, along lines already laid down at its commencement, rather than productive of new lines of thought or endeavor. I do not know that the record of the past has been surpassed either in magnitude of unit produced or in efficiency of performance. There has been some difficulty in

the production of large cylinders which would maintain their integrity under the extreme conditions of pressure and temperature which obtain when the entire process of conversion from the potential energy of the fuel to the energy of a revolving loaded shaft is conducted within the cylinder itself; and a metal which can be cast, and which has the strength to withstand the pressure in thicknesses which will allow the walls to be effectually jacketed, is a desideratum. Cast steel has gone far toward solving the problem. An alternative is a modification of design which shall permit the use of wrought metals.

Mr. E. T. Adams said early in the year that we were nearer a 10,000-h.p. unit, looking forward, than to a 5000-h.p. unit, looking backward. He has designed a station of 100,000-kw. capacity, which with 5000-kw. units will take no more space than a station of equal capacity with 14,000-kw. turbines, and we hope to have the particulars of the design for an early meeting.

The gas plant has been handicapped by the excessive first cost natural to a period of evolution and rapid development, but with standardization of design there will be a material decrease of cost. Present estimates would indicate a relation of first cost as between steam and gas power plants of about 100 to 130. The use of high piston speeds has done much to decrease the cost of the gas engine per unit of capacity, speeds of 1000 ft. per min. being not uncommon.

In Northern latitudes, where heating is an important factor, or in factories where large amounts of heat are used in manufacturing processes, the internal combustion engine is at a disadvantge because its efficiency is surpassed by that of the steam engine, when the latter is credited with its rejected heat available for such uses. The rejected heat of the gas engine, less than that of the steam engine by reason of its greater efficiency, is still some 75 per cent of that supplied to it. A large amount of this heat comes out in the jacket water at a temperature of about 140 to 180 deg.; the rest in the exhaust gases at a temperature of upwards of 1000 deg. A practicable method of applying this heat to useful purposes will greatly increase the field of the internal-combustion engine.

A number of engineers are at work upon the problem. One manufacturer passes such gases through conduits beneath the cement floor of his shop, maintaining the room at a comfortable temperature. Another inventor uses the exhaust gases to make steam from the already heated jacket water, supplementing them when necessary by burning gases from the producer beneath the

boiler. He labors under the disadvantage that the returns from the heating and process systems are not cool enough to go into the jacket and make the process a closed heat circuit.

The year has witnessed continued and new attempts at a gas turbine, with no results about which we can talk in detail. Mr. Hans Holzworth, one of whose steam turbines was exhibited at the St. Louis exposition, has exhibited a small gas turbine which ran so satisfactorily that he had no trouble in obtaining capital for the building of a 1000-h.p. unit, upon which he is now engaged at Berlin. Mr. W. A. Warman, one of our New York members, is obtaining some very interesting results in his efforts to construct a Hero turbine, Avery engine or Barker's mill, operated by gasolene.

The difficulty in the gas turbine is, of course, to find materials which can deal at the same time with pressures and temperatures both high. In view of the remarkable results obtained by the addition of low-pressure steam turbines to reciprocating engines, the conception of a gas turbine in series with a reciprocating gas engine naturally arises. The terminal pressure and temperature in a gas engine are high. There is considerable energy to be had by expanding the gas to temperatures attainable even under atmospheric pressures. At the same time, these gases are not so hot when they come from the engine that they cannot be readily worked in nozzles of easily procurable material. It would be interesting to see this adaptation worked out, at least on paper.

Notwithstanding the attractiveness of the two-stroke cycle, the smaller weight of engine required, etc., no progress has been made during the past year except upon engines of the smaller sizes, and we appear to be no nearer to the wider use of the two-cycle engine in large sizes than we were two years ago.

The movement toward the better use of by-product gases goes on. All the large iron and steel works use their blast-furnace gases in gas-driven blowing engines, pumps and electric generators, and coke manufacturers are putting in by-product ovens. It is said that German interests will erect a by-product plant near South Bethlehem which involves an investment of some $4,000,000 to furnish coke to the Bethlehem Steel Company, and which will have available some 24,000,000 cu. ft. of gas per day, of a thermal value of 400 B.t.u. At 10,000 B.t.u. per h.p.-hr. this amounts to 40,000 h.p. continuously.

Efforts continue toward a producer which will satisfactorily and continuously handle the more abundant and cheaper bituminous coals without expensive auxiliary cleaning apparatus and if reported

performances hold good, the year may be credited with notable progress in this direction.

Increasing interest is being taken in the use of substitutes for coal and oil. Numerous peat bogs are being worked and both peat and lignite are successfully used in producers. This is especially true of Western lignite. Occasional references are made to the possibilities of using town waste in the same way, but we have not learned of any notable progress in this direction during the year, and it is still a question whether the vast carbonaceous rejecta of our large cities can best be made to contribute to our fund of available energy by way of the producer or of the still.

The United States Geological Survey has just issued a bulletin, bringing the question of alcohol as fuel down to date. Burned in engines especially constructed for it, Messrs. Strong and Fernald, authors of the bulletin, say that a gallon of alcohol will develop as much power as a gallon of gasolene, notwithstanding it has but 71,900 heat units per gallon, as against 115,800 for the gasolene. The present price of denatured alcohol is 50 cents per gallon in five and ten-gallon lots, more than twice that of gasolene; but untrammeled by an internal revenue tax it would doubtless be a serious competitor of the latter fuel.

The usefulness of the gas engine in marine work was considered by Mr. Straub in his paper presented to the Section at the Washington meeting. Rumors persist to the effect that a sizable war vessel is being built in Great Britain, to be propelled by internal combustion-motors, and we understand that a contract has been awarded for a 1000-h.p. outfit for one of our own lake steamers. The successful application of gas engines to the propulsion of canal boats in Germany is well known.

REPORT OF GAS POWER RESEARCH COMMITTEE

A few months ago the Executive Committee of the Gas Power Section, recognizing the importance of a well-considered plan for the solution of the problems developed by the increasing use of gas power, appointed a Research Committee consisting of Prof. Robert H. Fernald, Prof. L. P. Breckenridge, Prof. Rolla C. Carpenter, Dr. Chas. E. Lucke, Prof. W. D. Ennis, Prof. W. T. Magruder, Prof. Harry N. Davis, Prof. Lionel S. Marks, Prof. David C. Gallup and Prof. W. H. Kavanaugh, with instructions to advise as to the proper lines of investigation to be conducted.

This committee, with the secretary of the Section as secretary, has prepared a list of the special problems in connection with the use of gas power, the solutions of which are urgently needed. It has been suggested that the well equipped engineering laboratories of colleges and technical institutions would be glad to coöperate in this work, and it is hoped that many of these problems will be taken up by laboratories and investigators and the results made public through The Journal of the Society.

A LIST OF PROBLEMS, THE SOLUTION OF WHICH IS DESIRED

A Fuels:

1 Compile from reliable sources comprehensive data concerning the coals of the U. S. which are worthy of consideration for use in gas producers .... L. P. B.

2 Use in the engine of producer gas direct from the generator with its full charge of tar .... R. H. F.

3 Use of bituminous coals and other tarry fuels in suction gas producers. R. H. F.

4 Use of briquetted lignite in producer. (Some work has been done at the Fuel-Testing Plant, Pittsburg.) .... W. H. K.

5 Sage brush, wood, in the producer .... R. H. F.

6 Use of peat in gas producers and methods of preparing it for use. R. H. F., W. H. K.

7 Use of briquetted peat in producer........ W. H. K.

8 Investigation of the rate (thoroughness) of dissociation of CO_2 in the producer at various temperatures .... W. D. E.

9 Ultimate analysis of ash in producer gas plants. (Also melting point of ash (clinkering) ....R. H. F., W. D. E.

10 Investigation of the temperatures prevailing in fuel beds of gas producers in conjunction with chemical investigations, R. H. F., L. S. M.

11 Ratio of air to fuel for best efficiency and comparison with theory. W. T. M., W. D. E.

12 Investigation of the effect on economic performances of varying the ratio of combustion.
- (*a*) In different types of producers with same fuel.
- (*b*) In same producer with varying sizes of fuel.
- (*c*) In same producer with varying mixture of fuel.
- (*d*) In same producer with coal mixed with limestone when the ash content of the coal is high..........................L. P. B.

13 Effect of depth of fuel bed in gas producer work............R. H. F.

14 Effect of sizing coal on producer efficiency...............R. H. F.

15 Determination of the effect of preheating the air in gas producers. R. H. F.

16 Determination of the effect of varying the amount of moisture in the suction producer ....R. H. F.

17 Quantity of steam required per pound of fuel in producer gas plants. R. H. F.

18 Best rate of burning fuel in various types of gas producers..R. H. F.

19 Effect of variable loads in producer work..................R. H. F.

20 Possible utilization and value of various kinds of crude oils in various types of internal-combustion engines ....R. H. F.

21 Relative values of alcohol and gasolenes as fuels for internal combustion engines ....R. H. F.

22 Latent heat of vaporization of gasolenes, kerosenes and distillates. W. T. M.

23 Apparatus for the gasification of tar in connection with producer plants ....W. H. K.

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33 Use of tar as a fuel in producers........................W. T. M.

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TESTING SUCTION GAS PRODUCERS

By C. M. Garland and A. P. Kratz, Published in The Journal for December

ABSTRACT OF PAPER

The paper describes a method of testing the suction gas producer which is independent of the engine. The engine is blanked off from the producer and a Schutte & Koerting steam ejector is inserted, which draws the gases from the producer and delivers them to a scrubber in which the steam used by the ejector is condensed. The gases then pass to a Westinghouse meter where the volume is determined.

A large part of the paper is devoted to the forms used in the computation and presentation of the results on gas-producer tests. Three forms are given, Nos. 1, 2 and 3. Form No. 1 is used only for the final presentation of the results of the tests; form No. 2 includes the results of all computations for convenience in computing; and form No. 3 contains the derivation and the discussion on the derivation of the formulæ used. The formulæ appearing in this form are arranged in the order of computation and the item numbers refer to the items of form No. 2.

The results of one test are included in the paper together with a graphical log illustrating the conditions during this test.

DISCUSSION

Prof. R. H. Fernald. In connection with the Government investigations, the feeling has prevailed that ever since the beginning of the work in 1904, gas producers could be tested on practically the same basis as steam boilers, i. e., without necessarily operating an engine in connection with the test. This would mean discharging the gas into the air in a manner similar to the discharge of steam in boiler test practice. This method of procedure has not been adopted at the Government testing station because so much prejudice has existed against the gas producer and gas engine. It has therefore been necessary that the gas generated at the testing station be utilized in an engine in order to avoid any discussion relating to the uncertainty of such operation. This has been particularly necessary owing to the large variety of fuels that have been handled and the variation

in the quality of gas produced. It is true, however, that from the producer standpoint alone the engine is not essential, and the method suggested by Mr. Garland is ingenious and reasonably convenient.

2 There are a few points in connection with this paper upon which further information is desirable. In Par. 5 it is stated that the weight of steam was measured by passing the jet through a calibrated orifice in a thin plate. Methods of determining the quantity of steam used by gas producers seem to be varied and the results obtained somewhat uncertain. I believe it would be interesting to know the details of the method employed by Mr. Garland. In the testing station at St. Louis the steam used by the pressure producer was determined by means of a calibrated orifice, but the fluctuations in pressure were such that the readings obtained were not regarded as absolutely reliable. During tests covering a period of approximately two years the steam used varied from 0.28 lb. per lb. of coal fired to 1.13 lb. of steam per lb. of coal fired. The average for twenty consecutive tests showed 0.69 lb. of steam per lb. of coal fired. It should be borne in mind that the fuels used for the different tests were quite different in composition and that the amount of steam required by the different fuels may have varied considerably; but in spite of this fact the feeling which prevailed about the plant was that the method of determining steam by means of calibrated orifices was not entirely satisfactory unless the pressure of the steam passing into the producer, and the percentage of moisture in the steam, could be kept constant during the test.

3 At the Norfolk station, however, the steam required by the producer was supplied by an auxiliary boiler, so that all water passing into this boiler could be positively measured. Although the coals used for the six tests reported below were practically the same in composition, yet the records show the steam consumption per pound of coal fired to be decidedly variable, as follows:

(1) 1.12 lb. per lb. of coal fired	(4) 0.82 lb. per lb. of coal fired
(2) 1.14 " " " " " "	(5) 0.77 " " " " " "
(3) 1.04 " " " " " "	(6) 0.69 " " " " " "

This wide variation shown for these six tests is due entirely to the methods of operation, and not to uncertainties in measurement, as might at first be inferred. There is need of systematic and careful investigations relating to this question of steam per pound of fuel. At the Pittsburg station the method of determining the amount of steam used in the vaporizer is by means of a water tank calibrated in pounds, thus insuring accurate measurement.

4 In Par. 6 is presented the general method of determining the amount of fuel used. One phrase attracts especial attention: "at the end of the test the fuel bed being brought to as near the starting condition as possible." In boiler practice where the quantity of fuel on the grate at any one time is relatively small, it is undoubtedly possible, within a reasonable percentage of error, to determine the condition of the fuel bed and to make this condition practically the same at the beginning and close of an eight or ten-hour test.

5 However, the situation is totally different in gas-producer practice in which the initial fuel supply and the amount of fuel on the grate at any given time is large compared with the amount required by the plant during a run of a few hours only. Even though the conditions at the close of a producer test be made to duplicate those at the beginning, there is still considerable difficulty in determining the exact fuel consumption, owing to the lack of accuracy in determining the true thickness of the fuel bed. In a producer of 250 h.p. rating it is not uncommon to make an error of from four to six inches in the true depth of the fuel bed. In a producer of this size, this will cause an error of about 800 lb. of coal, or about 400 lb. of coke, according to the condition of the fuel bed at the time. It is imperative, therefore, that the tests of producer plants be continued to such length that these errors in measurement will be but a small percentage of the total fuel consumed.

6 Mr. Garland states that it was endeavored to make the tests of such duration as to bring the probable error of filling down to about two or three per cent. It will be of interest to have explained in further detail the method of procedure used in determining the exact amount of fuel consumed. With a 250-h.p. plant in which the fuel consumption for a period of 8 hr. amounts to only 1800 lb. approximately, the error due to inaccurate measurement of the depth of bed and variations in fuel bed thickness may be as great as 1150 lb. The percentage of possible error in calculating fuel consumption for short periods is obviously great. With a period of 24 hr. and a fuel consumption of about 5400 lb., the percentage of possible error, although much less, is still over 20 per cent.

7 In the producer tested the effective fuel bed volume was approximately 4 cu. ft., which is equivalent roughly to 250 lb. of anthracite pea coal. It is probable that a large percentage of the gas value of this coal may be given off without materially decreasing the fuel volume, under certain conditions of fuel bed. In a run in which the fuel consumption for this producer amounts to only 800 lb. with an

initial bed of 250 lb., it is a question whether the percentage of error in fuel bed estimates may not amount to 10 or 12 per cent instead of 2 or 3 per cent.

8 In a recent paper on this subject published by the United States Geological Survey, the following conclusions were presented:

a Throughout a test the fuel bed should be maintained in uniform condition, with regard both to character of the fire and thickness of the bed.

b Failing in this, special care should be exercised to see that the fuel bed is in the same condition and of the same thickness at the close of the complete test or at the end of a test period, as at the beginning.

c A test should never be started when the producer has been standing idle for some time with banked fires, as the fuel bed will not be in the average condition under which it will be required to work during the test.

d If, as the appointed hour for closing the test approaches, the fuel bed is not in the proper condition, the time of ending the test should be postponed until the bed naturally assumes the proper thickness and character. No forcing of conditions should be allowed simply to bring the test to an end at a previously determined hour.

9 In Par. 12 it is suggested that the volume of gas may be computed from the analyses of the gas and coal and the statement is made that this "may be relied upon within 5 per cent, provided the sampling is accurate." This last clause "providing the sampling is accurate" seems to contain the essential point. Time does not permit a lengthy discussion of this important subject, but too much emphasis cannot be placed upon the fact that proper sampling is difficult to accomplish.

10 Reference to the packing of the ash in the fuel bed suggests another point which must be very carefully considered in making fuel bed measurements, viz., the swelling of any coals due to the application of heat. Frequently in our government tests, the measurements of the fuel bed have caused very misleading impressions due to the fact that the fuel had swollen materially during the operation of the plant.

11 In Form 1 a number of items appear under a heading "Quantity of Air." Although it is quite possible to determine small quantities of air with some degree of reliability, yet methods for making such measurements of large quantities appear to be entirely lacking.

Further details of the methods pursued in this test will, I believe, prove of interest.

12 In items 128 and 128*b*, are presented the producer efficiencies based on dry coal and combustible. It is not apparent why there should be a difference of 4 per cent in the efficiencies shown.

G. M. S. Tait. The usual method of testing a plant for such a short period would be to operate the producer for two or threedays beforehand so as to bring the fuel bed to an average working condition, that is, with an average amount of carbon in proportion to ash. Then a comparatively short run, provided great care was taken as to the fuel depth, would give fairly reliable figures. Otherwise, when drawing on a fresh fire and making a run of only twelve hours, it would be necessary to pull the entire fuel bed at the end of the run and analyze the contents for carbon and ash, in order that any sort of accuracy might be obtained.

2 In one of the author's tests, instead of 34 lb. of coal per sq. ft. of grate area, 8 to 10 lb. would be a normal figure, as 34 lb. of coal per sq. ft. is entirely impracticable for anything but a very short run on American fuels. In this test a large part of the coal originally in the producer was apparently burned to ash, and its consumption was completely left out of the test, causing very erroneous results.

3 Attention is called to the fact that the ash content in the ashpit is practically much less than the ash content of the fuel, as shown by analysis. The balance of the ash is undoubtedly in the fuel bed and its presence there entirely upsets the basis of calculation for this paper. It is safe to say that a two days' run would have given a reversal of the first day's figures.

H. H. Suplee. I would like to speak of the unreliability of an orifice as a means of measuring. Only this morning a member called my attention to the discharge of steam from a boiler in which the orifice and all conditions surrounding it were identical in several tests. The amount of steam generated was measured by carefully weighing the water, double-checking it in tanks, and yet there was a variation of ten to fifteen per cent in the results, the steam pressure and the temperature being kept as uniform as possible. This fact casts a doubt on the orifice as a means of determining flow.

L. B. Lent. The figures given show that the draft through the producer was practically 1½ in., and yet 38.8 lb. of dry coal was

burned per sq. ft. of grate area. Still, with this consumption the producer efficiency seems to be very good. My impression is that this is a remarkable rate of consumption in producers of large type; and I would like to know if this is the common practice in smaller sizes of producers.

H. F. Smith. Regarding the conditions of the fuel already discussed it seems to me that the author has presented all the necessary evidence to show that the conditions in the fuel bed were not the same at the end as at the beginning of the test.

2 In the graphical log in Fig. 3, it will be noticed that the temperature of the gas leaving the producer at the beginning of the run was 400 deg. fahr., and at the end of the run something over 1300 deg. fahr. The rates of gas production and fuel consumption were practically uniform. It is evident that there was some variation in conditions, otherwise this difference in temperature would not have occurred.

W. B. Chapman. Perhaps I can answer Mr. Lent's question in regard to the quantity of coal gasified in producers. Producers for furnace work are usually rated at 10 lb. per sq. ft. of internal diameter on Pennsylvania coal, but only 7 lb. per sq. ft. on Illinois coal. The best record I have seen for *hand-operated* bituminous coal producers was 16 lb. per sq. ft. Mechanically agitated producers gasify from 15 lb. to 30 lb. per sq. ft.

2 The question of the amount of anthracite coal that can be gasified is very interesting. Engineers from abroad say that two or three times as much can be gasified as is the custom in this country. Every gas producer manufacturer in this country having a foreign engineer in charge has designed his first producer very much too small. The more experienced manufacturers do not rate their producers at more than 10 lb. per sq. ft.

3 It is strange that we cannot get the results said to be obtained in foreign countries. The difference must be in the coal.

Prof. R. H. Fernald. In reference to the rate of burning per square foot of grate area, I desire to call attention to the high figures shown by Mr. Garland. These figures seem to be very unusual for this type of producer even under the test conditions described. The highest rate with which I am familiar in commercial operation is that found in the case of a large installation using lignite as fuel. This

plant shows a daily rate of 33 lb. per sq. ft. of fuel bed area per hour during 16 hours each day and 48 lb. during the remaining 8 hours. In this installation the producers are of the down-draft type, but even under these conditions this rate is, I believe, exceptional.

2 In reference to Mr. Chapman's remarks about the manufacturers abroad, I would say that apparently all of them stipulate the type of coal that shall be used in their producer. They specify that the coal must be of such and such a grade, non-caking and with only such and such a percentage of ash and tar. As nearly as I was able to ascertain, practically every manufacturer abroad has reached the conclusion that it is wise to designate definitely the coal to be used.

3 In one suction producer in Germany, operating on bituminous coal, I found that the successful manipulation of the plant was due to the fact that three kinds of coal, mixed in the proper proportions, were being used. In other words, this type of producer using bituminous coal as fuel was entirely feasible in the home plant of the manufacturer, but it would hardly prove a saleable article in this country, as it would be almost impossible to guarantee the three required grades of coal at all times. It would also be out of the question to secure operators at a reasonable compensation who would give the plant the required attention.

E. N. Trump. The rate of combustion in producers using anthracite coal depends very much upon the size of the coal. Seven tons per 24 hours, with a producer 7 ft. in diameter, is about the maximum for No. 1 buckwheat coal. This equals 15 lb. per sq. ft. of grate surface per hour.

2 Burning Western coals in producers, especially Hocking Valley coal, a high rate of combustion is obtained. I have operated producers continuously for a considerable period at the rate of 42 lb. per sq. ft. of grate surface. This is with a large percentage of steam in the air, also with mechanical ash extraction, the fuel bed being thus kept well agitated.

2 Venturi meters give very accurate results in the measurement of both gas and steam, much more accurate than the simple orifice.

The Authors. It will be well to emphasize the fact that the producer under discussion was designed and intended for intermittent service only; that is, it is not suitable for runs of greater than 12 to 18 hours duration. This is due to the small size of the producer, and

the absence of charging bell, water-sealed ashpit and mechanical means for agitating the fuel bed.

2 Owing to the small size of the producer and the absence of means for thoroughly cleaning the fuel bed from time to time, as above noted, the accumulation of ash toward the end of 12 or 15 hours of continuous operation is so great as to necessitate such thorough cleaning as seriously to lower the heating value of the gas.

3 From the foregoing it will be evident that our test corresponds to the conditions under which the producer is normally operated. Owing to the thorough cleaning of the fires before starting the test, and the removal of the ash from the grate, a large quantity of green fuel is brought into the path of the outgoing gases, resulting in their being cooled. At this time, the temperature of the fuel bed is also lower, as indicated by the analysis of the gases over the first two hours of the test. The heating value of the gas is not lowered, for two reasons: first, the descent of the green fuel into the path of the gases results in the distillation of the CH_4 and other heavy hydrocarbons; secondly, an increase in the percentage of hydrogen results from the lower temperature of the fuel bed.

4 At the close of the test the fuel bed was evidently at a higher temperature than at the start. This resulted in increasing the unaccounted-for loss in the heat balance, but its extent (estimated from the results of a number of tests) is about one per cent for the present test. This, it is believed, explains the condition pointed out by Mr. Smith.

5 Professor Fernald and Mr. Tait call attention to the probable inaccuracy in determining the weight of coal fired on the test. We have recognized this source of error, and in Form 2 have included such items as give proof of the accuracy of the work through the stoichiometric relations. As the full import of these items has evidently not been realized, we will amplify them.

6 First, to determine approximately the purely mechanical error in estimating the weight of coal fired during the present tests, the producer was filled four separate times, and the weight of coal required was noted in each case. The average of the four weights was taken as the mean or true weight of coal required to fill the producer. The results are given in Table 1, herewith. It will be seen from this table that the maximum variation from the mean is 8.75 lb., or 1.7 per cent. This in the test under consideration represents an error of probably 1.1 per cent.

7 Mr. Tait seems to think that the presence of the ash in the fuel bed "upsets the basis of calculation for this paper." The total weight of ash in the dry coal is 776.5 lb.; 13.17 per cent = 102 lb., of which 52 lb. was taken out in the ash and refuse, leaving 50 lb. remaining in the fuel bed. This would seem to indicate an error of 6.3 per cent, due to failure to remove this ash. Since the ash is soft and fine it would pack into the interstices between the coals so that its volume would by no means displace the same volume of coal. If it displaced one-half its volume of coal it would cause an error of slightly over 3 per cent. It is probable that its presence caused even less error than this. In order to bring out the different errors we will analyze the conditions existing on the test.

TABLE 1 WEIGHT OF GREEN COAL REQUIRED TO FILL THE PRODUCER

Trial Number	Weight Lbs.	Variation from Average Weight	Per Cent Variation
1	669.25	−8.75	1.70
2	676.25	−1.75	0.26
3	683.25	+5.25	0.77
4	683.25	+5.25	0.77
Total	2712.00		
Average	678.00		

8 It is probable that the composition of the producer gas on leaving the scrubber, and at any two points in the cross section of the main, is the same. In order to eliminate such an uncertainty, however, we have taken the gas for our samples simultaneously from different points in the cross section of the main and at a point beyond the scrubber, by means of the sampling tube illustrated in the paper. These samples were taken continuously over the period of the test, both for analysis and for the calorimeter. The heating value of the gas, as computed from analysis, is 138.1 B.t.u. After corrections were made for the error in the meter, the error due to the vapor pressure of water, and the error due to radiation and conduction into the calorimeter, the heating value of the gas as determined by the Junker calorimeter was 137.3. Since the heating value as determined from two separate samples of gas, by two independent methods, and by two independent observers, checks within 1 per cent, it must be admitted that the sampling, the analysis and the heating value of the gas are probably correct within less than 1 per cent.

9 The volume of gas generated by the producer was measured by

a Westinghouse meter, guaranteed by the company to be accurate within 2 per cent. However, as a further precaution, the meter was carefully recalibrated and was found to be accurate within this limit. A calibration curve was plotted from the calibration, so that the error in determining the gas volume must have been within 2 per cent, and was doubtless even closer than this.

10 As shown by a number of tests on the present fuel, the coal was fairly uniform. A sample representing about 15 per cent of the coal fired was mixed and quartered until about eight or ten quarts remained. This was then ground, and again mixed and quartered until sufficient to fill a quart jar remained. The heating value from this sample as determined by the calorimeter was 13,040 B.t.u. per lb. The mean of eight determinations on this same fuel showed a heating value of 12,900 B.t.u. The probable error in the analysis and in sampling the fuel, judging from the heating value, is doubtless not greater than 1 or 2 per cent.

11 We have noted the volume of gas computed from the analysis of the coal and the analysis of the gases in Form 2, Item 126. This volume is 56,200 cu. ft. of standard gas, while the volume as actually measured by the meter, corrected for the vapor pressure of water, is 57,500, showing a discrepancy of about 2.3 per cent. The volume determined from computation was obtained from the formula of Item 126, Form 3. It is based on the fact that the weight of carbon in the coal fed to the producer must equal the weight of carbon appearing in the producer gas, plus the carbon lost in the ash, plus the carbon lost in soot and tar, plus the carbon lost by the absorption of CO_2 and CO by the scrubber water. The carbon lost in the ash is readily obtained, the carbon lost in the soot and tar is not over 1 per cent, while the carbon lost through the absorption of the gases by the scrubber water is also very small.

12 It may be well to compute the carbon in the gas and compare this with the carbon fed to the producer in the coal. We will compute the latter first. The total carbon in the coal is 0.7984 x 776.5 = 620 lb. The total carbon in the ash is 0.388 x 85 = 33 lb. The carbon that should appear in the gas is therefore 587 lb. The total weight of gas from Item 131, Form 2, is 3912 lb.

Carbon in CO_2 of gas	$= 0.0716 \times 12/44 \times 3912$	$= 76.4$
" " CO " "	$= 0.2925 \times 12/28 \times 3912$	$= 490.5$
" " CH_4 " "	$= 0.0112 \times 3/4 \times 3912$	$= 32.8$
		599.7

Thus 599.7 lb. is the total weight of carbon appearing in the gas as measured by the meter. 599.7 − 587 = 12.7 lb. of carbon unaccounted for $= \frac{12.7 \times 100}{599.7} = 2.1$ per cent. As already stated, there may be an error of 1 per cent in the meter by which the above volume of gas was determined, the error being either positive or negative. There may have been 1 per cent of carbon lost in the soot and tar, but not more than this; there may also have been an error in the analysis of the coal amounting to $1\frac{1}{2}$ per cent. We estimate the principal errors in the test as follows:

	%
Error in filling the producer	−2.1
Gas analysis or heating value of gas	±0.7
Volume by meter	±1.5
Coal analysis and sampling of coal	±1.5
Carbon lost in soot and tar	−1.0
Loss in sensible heat in the fuel bed due to the lower temperature at the start than at the close of the test	−1.0

The total error in the results of the test that would affect the cold-gas efficiency of the producer, if all the above errors are assumed as accumulative, equals .8. The probable error is 2.7.

13 There are three other errors that may affect the heat balance, namely, the error in measuring the temperature of the outgoing gases, the error in the determination of the specific heat of these gases, and the error in the amount of steam fed to the producer. The error in measuring the temperature of the gases may be 2 per cent; the error in determining the specific heat of the gases may be 6 per cent; the error in determining the steam fed to the producer may be 25 per cent. If these errors are accumulative, the first two represent a total error based on the heating value of the fuel of about 2 per cent, and the third of about 1 per cent. Therefore, if all errors are accumulative, the total error in the heat balance is about 6.8 per cent; as some of these errors will be positive and others negative, the probable error in the heat balance is about 3.5 per cent. As the heat balance shows an unaccounted-for loss of 4.4 per cent, about 1 per cent being radiation and conduction, the actual error in measuring the coal delivered to the producer on this test could not have exceeded 3 per cent. We have therefore been able to run tests of such duration as to reduce the probable error in filling the producer to 2 or 3 per cent. Furthermore we believe the results indicate that they are above the average in

accuracy for this kind of work, as we have seen very few tests on producers that would stand the above analysis.

14 As Professor Fernald and Mr. Suplee have pointed out, we have found the use of the thin plate orifice for the measurement of steam not altogether satisfactory. As the heat supplied in the steam on most of our tests is small, a large error is permissible in the measurement. Our aim has been to vary the pressure on the orifice so as to keep the hydrogen content of the gas practically constant. It might be well to state that the orifice was used only while we were obtaining a new vaporizer for the producer.

15 We have found no tendency in the anthracite coal to swell. We believe that this is a property of bituminous coal containing large quantities of moisture and of hydrocarbons.

16 As the quantity of air does not enter into the computation of the more important quantities, it was computed from the nitrogen in the producer gas. The formulae for this computation are given in Form 3.

17 The difference in the efficiency based on dry coal and the efficiency based on combustible, as noted by Professor Fernald, is due to the fact that we have used the word combustible as defined in Form 3, Item 54. This takes into account the grate efficiency. The result is that the efficiency based on combustible corresponds to the efficiency based on 100 per cent grate efficiency. It is used for the reason that it is often desirable to show relations between efficiency and other quantities that are independent of the grate efficiency.

18 The amount of coal burned per square foot of grate area is a very variable quantity and depends upon the size of the fuel, the kind of fuel, the nature of the ash, the amount of water supplied, the proportions of the producer, the operation and the length of run.

19 For intermittent work, such as the present producer is adapted for, and with coals containing an ash infusible at temperatures under 2300 deg. fahr., it is possible to operate at several times the capacity possible with coal containing a fusible ash which necessitates a low fuel bed temperature.

20 The rapidity and extent of the reaction of CO_2 on incandescent carbon depend upon the temperature and upon the catalytic action of the fuel. At a given temperature and an indefinite time of contact of gases with the incandescent carbon, a definite amount of CO_2 and CO will be formed. The lower the temperature the less the per cent of CO formed and the longer the time required for equilibrium, so that with low temperature in the fuel bed the time of contact of the

gases with the fuel must be greatly increased. The same is true for the reaction of water on incandescent carbon. Harries[1] passed water vapor over incandescent carbon at different temperatures and obtained the results given in Table 2. These results show the effect of temperature upon the water-gas reaction. Due to the low temperature, the CO_2 is high, the CO is low and the ratio of water decomposed to water supplied is small. The latter fact, in the case of the producer, results in lowering the efficiency, as the undecomposed water carries out a large quantity of heat.

21 The curves of Fig. 1, herewith, taken from Dr. Clements'[2] work on the rate of formation of CO in gas producers, illustrates the effect of the time of contact, expressed in terms of velocity in feet per second, upon the amount of CO formed in passing CO_2 over incandescent anthracite coal. At a temperature of 1100 deg. cent., and a time of contact corresponding to a velocity of 1 ft. per sec., 11 per cent of CO is formed. If the velocity is reduced to 0.1 ft. per sec., so that the time of contact is increased ten times, 70 per cent of CO is formed. If an indefinite time of contact is assumed, equilibrium is reached at

TABLE 2 EFFECT OF TEMPERATURE ON WATER-GAS REACTION

Temperature Deg. Cent.	H_2	CO	CO_2	H_2O
674	8.41	0.63	3.84	87.12
838	28.68	6.04	11.29	54.09
954	44.43	32.70	5.66	17.21
1125	50.73	48.34	0.6	0.303

this temperature with 90 per cent of CO formed. This illustrates why it is necessary to use a small rate of combustion per square foot of grate area, due to operating with coals requiring a low temperature for the prevention of clinker formation.

22 If in the example just cited the temperature had been 1300 deg. cent. in the fuel bed, 70 per cent of CO would have been formed at a velocity of 0.5 ft. per sec. The time of contact would have been reduced five times, so that the rate of combustion could have been increased almost five times without appreciably changing the composition of the gas or the depth of the fuel bed.

23 In the case of our tests with the Scranton pea coal, we have

[1] Habers, Thermo-dynamics of Technical Gas Reaction, p. 138.

[2] Bulletin No. 30, Engineering Experiment Station, University of Illinois.

been able to vary the coal per sq. ft. of grate area from about 10 lb. to 45 lb., without appreciably affecting the efficiency of the producer. At the higher rates of combustion, however, the producer requires much more attention. If it were not for the fusion of the ash, the weight of coal per square foot of grate area could be increased indefinitely by the use of a blast and a sufficiently deep fuel bed.

24 The term "coal per square foot of grate area," as used in producer practice, is not, we believe, a true basis of comparison for the operation of different producers, for the reason that the coal per

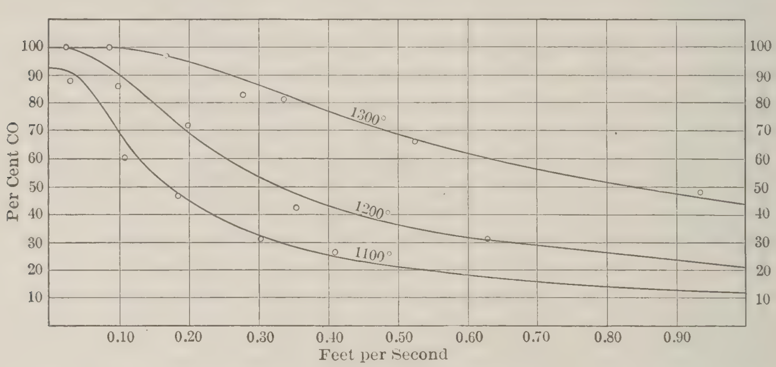

Fig. 1 Velocity of Gas in Feet per Second, Fuel Bed 1 ft. Deep

square foot of grate area depends to a certain extent upon the depth of the fuel bed. For this reason, largely, we have used a term, "rate of descent of coal through the fuel bed," or "coal per cubic foot of fuel bed per hour," which appears under Items 70 and 71 in Form I.

BITUMINOUS GAS PRODUCERS

By J. R. Bibbins, Published in The Journal for December 1909

ABSTRACT OF PAPER

This paper attempts to throw some light on the results of the development of a comparatively new type of apparatus, the double zone bituminous gas producer.

Much time and money have been spent by the various manufacturers in the development of a tar-free gas producer and in some respects the obstacles have seemed insuperable. Every advance is therefore of interest and importance, and it has seemed worth while to report a long series of tests conducted by the builder to determine the net results under commercial conditions, whether good or bad. These tests are characterized by their unusual duration and absence of outside conditions affecting the results.

These results in general will speak for themselves and it is necessary simply to emphasize the fact that continuous operation has been secured with tar-free gas of reasonable heat value and producer efficiency and an over-all plant economy of about one pound of fair bituminous coal per brake horsepower (proportionate economies for poorer grades). More important still, the fact has been developed that the efficiency and general effectiveness of operation of the producer on low grade fuel, lignites, etc., is practically as high as with the higher grades. This places within the reach of the producer the enormous fuel deposits of the West and South, which are practically invaluable for steam work.

DISCUSSION

G. M. S. Tait. The results reported in this paper are entirely in accord with what we have found, namely, that the gas of the lesser British thermal units is much more satisfactory for engine practice. In other words, the efficiency of a gas of 90 B.t.u. is proportionately double that of a gas containing 600 B.t.u. per cu. ft., the gases in question being respectively blast-furnace gas and gasolene vapor.

2 I would like an expression of opinion as to the reason for this great discrepancy in efficiency between the two gases, my own opinion being that the excessive normal losses are due to the sudden high temperature developed in the gas of high B.t.u., which is greater than can be handled by normal piston speeds.

3 The tar washer used in this test appears to be a succession of water seals and I would like to know what would be the total frictional effect of these seals under normal conditions and on full load.

4 In all producers properly designed the thermal efficiency appears to remain constant between 20 per cent and 100 per cent load. I can confirm Mr. Bibbins' experience as to the action of this particular class of fuel and its desirability for producer work.

Prof. R. H. Fernald. Mr. Bibbins places as his first essential requirement "continuous operation 365 days per year," and states that any departure from this condition means reserve equipment. He also states that the condition for producer operation must parallel steam boiler practice.

2 It is undoubtedly true that a producer which will operate continuously 365 days a year would prove a splendid commercial proposition, but it seems to me that in the requirements outlined the conditions imposed are much higher than those of any steam boiler plant and are beyond practical requirements. Every plant of any size must necessarily have one or more reserve units, as no plant can operate continuously 24 hours a day 365 days a year. If the producer described by Mr. Bibbins can approach this operating condition, it will certainly revolutionize our present day power-plant practice. It would seem advisable, in the light of the present development of gas producers, to impose conditions which are less severe.

3 Relating to the adaptability of a single producer to all classes of fuel, it is well to bear in mind that the government testing station has practically demonstrated the fact that almost any variety and grade of our recognized fuels can be handled with more or less success in a given producer installation without change of details of design. It is questionable, however, whether such practice lends itself to the efficient use of a wide range of fuels. It is probable that better results can be obtained by utilizing a producer type to cover a certain range or variety of fuels and another plant of somewhat modified design for another range.

4 Mr. Bibbins refers to the excessive labor required by most producers. At the present time the labor requirements are excessive for the majority of the plants utilizing bituminous coal. This labor, however, even under bad conditions of operation, such as those involved when the fuel is one that clinkers badly, probably does not exceed that of the average steam installation, although the labor is of a somewhat different character. During the regular operating

period of the plant this labor may amount to very little; but at the close of a week, two weeks, or any length of operating period, in the commercial plants now in operation, cleaning may be an exceedingly dirty, hot and tedious operation. With the steam boiler plant the labor is more uniformly distributed. In spite of the more erratic and more violent labor required at times by the producer installation, the total cost for cleaning, ash removal, etc., is probably within the limits of the average steam installation.

5 Experience with a large variety of fuels leads one to question whether the treatment accorded one fuel in order to prevent clinkering will produce the same results with a fuel possessing totally different characteristics. The impression from the tests carried on at the Geological Survey testing station is that fuels varying greatly in composition and in characteristics require widely different treatment. This impression has been obtained from tests on a large variety of fuels, but the number of tests on each of the different fuels was not sufficiently large to warrant positive conclusions regarding this point. European practice, however, seems to confirm this opinion, as practically every producer manufacturer finds it imperative to specify coals of certain characteristics for use in his type of producer and does not guarantee the plant on fuels outside of this class.

6 In the discussion of the results the point is brought out that with Texas lignite the rate of combustion in this producer can be so increased as to permit the same rating of the producer as when operating on a high-grade fuel. Note is made of the fact that a charging rate of 27.2 lb. per sq. ft. per hr. was obtained with this lignite. An installation in Texas, which I visited a year ago, consisted at that time of three producer units of 1100 h.p. rating each, or a total of 3300 h.p.

7 Owing to the character and high percentage of the ash, together with the excessive demands upon the plant each unit was cleaned every third day, or, what amounts to the same thing, one unit was cut out of operation during a part of each 24-hr. day. It required eight hours to cut out the gas from a given unit, to clean thoroughly, rekindle fires and cut in the new gas. During each 24-hr. day, then, the full plant capacity, rated at 3300 producer h.p. was in operation 16 hr., while only 2200 producer h.p. were in operation the remaining 8 hours. During the entire 24-hr. period, however, according to the operating records, the engines were developing 2800 h.p. The operating records also showed that the fuel consumption per square foot of fuel bed area per hour amounted to 33 lb. during the 16-hr. period and 48 lb. during the 8-hr. period.

8 The statement is made that the economy of less than 1 lb. per b.h.p.-hr. is probably below previous results in bituminous producers. It is assumed that this statement is not intended to cover the tests at the government testing station, which has reported a number of instances in which the consumption varied between 0.8 lb. and 1 lb. per b.h.p.-hr.

9 Mr. Bibbins states that perhaps the most important result is tar-free gas. It is undoubtedly true that tar-free gas is eagerly sought in all cases in which the gas is to be used in engines. In my own mind, however, it is somewhat questionable whether tar-free gas, as reported in this paper, means that the gas from any and all fuels used in this plant would necessarily be free from tar. Experience with a producer of somewhat different design shows tar-free gas with the majority of fuels, but in the case of certain fuels the results are quite the reverse. If the producer under discussion can produce tar-free gas from any and all varieties of fuel, it is certainly a development in the right direction.

10 In the closing paragraph of Mr. Bibbins' paper the impression is conveyed that the steam boiler units of 2000 and 3000 h.p. are found not infrequently, and that producer units are small in com-

TABLE SHOWING CAPACITIES OF PRODUCER-GAS POWER PLANTS

	No. of plants	Horsepower				Per Cent of Total	
		Total	Average	Minimum	Maximum	No.	H. P.
Anthracite Coal:							
Over 500 h.p.	8	7,550	950	600	1500		
500 h.p. or less	407	40,550	100	15	500		
Total	415	48,100	116	15	1500	88	43
Bituminous Coal:							
Over 500 h.p.	20	49,000	2,450	750	6000		
500 h.p. or less	17	5,150	300	35	500		
Total	37	54,150	1,460	35	6000	8	49
Lignite:							
Over 500 h.p.	3	7,275	2430	525	3750		
500 h. p. or less	19	1,725	90	25	250		
Total	22	9,000	410	25	3750	4	8
All Plants	474	111,250	235	15	6000	100	100

parison with the usual boiler unit. In my opinion the condition at the present time is quite the reverse of this. In European practice it is not uncommon to find producer units of 1250 and 2500 h.p., and in the United States units of considerable size are in commercial operation, as shown by the accompanying summary of the producer-gas power plants operating in June 1909. There are undoubtedly over 500 plants in operation, as the list includes 474.

11 It is true that many of these larger plants are made up of several units, but an inspection of the original data shows the following single units of 500 h.p. or more:

H.P.	No.	H.P.	No.
500	4	1000	10
625	6	1500	1
750	3	2000	7

One single unit of 3,000 h.p. and one of 4,500 h. p. are reported, but these figures have not been verified.

12 It is interesting to observe that about 88 per cent of the total number of installations in the country are operating on anthracite coal (a few using charcoal or coke) and that bituminous coal and lignite are used in the remaining 12 per cent. It is not strange, therefore, that the majority of plants are at present made up of relatively small units, although the number of large units is rapidly increasing as bituminous plants are becoming more common. In point of size the bituminous plants at present average 12½ times the size of the anthracite plants. Of the total horsepower approximately 57 per cent is derived from bituminous coal and lignite, and 43 per cent from anthracite coal, charcoal and coke.

13 Although in large central stations there are many operating advantages in relatively small units, yet it is believed that in the near future central station development will demand equipment of much larger capacity. A consideration of the fuel resources of the country indicates that in order to keep the price of power developed from fuel down to a consistent figure

a Grades of fuel which warrant transportation, or which may be defined as "marketable," should be used with the greatest practical economy.

b The very large percentage of coal of so-called low grade which today is left at or in the mine must be utilized.

c Advantage must be taken of the large deposits of lignite and peat which are found in many sections of the country.

It is undoubtedly true that in general, under conditions which do

not require the use of steam for other than power purposes, the producer-gas power plant meets the requirements of *a*. At present the only method of advantageously handling the fuels mentioned in *b* and *c* is in the gas producer, and the utilization of these lower grades of fuel on an extensive scale demands concentration of large power units within close proximity to the fuel supply.

W. B. Chapman. In Par. 3, among the different requirements for successful operation, is mentioned the prevention of clinkers. I think the formation of clinkers can be avoided by the prevention of blow-holes or chimneys which allow the air to blow up through the fire bed, making hot spots. The average temperature across the hot zone in a producer is seldom high enough to produce clinkers. It is only in the neighborhood of the blow-holes that a sufficient temperature is attained to form clinkers. If the excessively high temperature necessary to the formation of clinkers existed throughout the producer, a clinker a foot or so thick would form immediately across its entire width. When ashes are melted they tend to run together, forming a clinker. The way to prevent this is to agitate the fuel bed continually, just enough so that the molten ash running down cannot take a permanent set in large masses, but is constantly kept in small pieces.

2 The successful producer should keep the fuel bed at an even temperature and uniform density throughout any horizontal plane. If there is a lesser density in any particular spot, the air blast immediately makes for this spot, causing an uneven temperature. To obtain this uniform density and temperature I believe that it is necessary to use some sort of mechanical agitation by hand methods, as no man or group of men can maintain a fuel bed of uniform density and temperature throughout any given horizontal plane long enough to get satisfactory results from soft coal.

3 Another point is that the successful producer should be made in a variety of sizes. The principles used in the producer described do not seem to admit of such variety. If this producer is of large diameter, the draft will go down the walls rather than in the middle, and the upper zone will not get hot enough in the middle to drive the tar out of the coal. If the tar is not removed by high heat in the upper zone, it is sure to get to the engine.

4 A successful producer should not require a delicately balanced draft, for the "balance" is often difficult to maintain. Uniform density in the two zones is imperative in double-zone or balanced-

draft operation, as otherwise the draft will vacillate from one zone to the other according to their varying density or resistance. The density is apt to change with the loads and with change of operators. The density will also change when the ashes are removed, as during this process a cavern is often formed which drops suddenly. In a producer of this type I have seen the vacuum vary from 2 in. to 18 in. in the lower or up-draft zone, and from 10 in. to 30 in. or more in the upper or down-draft zone.

5 In Par. 21, referring to the question of varying the air supply to the engine according to variations in the heat value of the gas, Mr. Bibbins says: "But this variable factor has received practically no attention and as a consequence producer operators are working entirely in the dark." To my mind the proper way of overcoming this difficulty would be to provide suitable mechanical means for maintaining uniform conditions in the organization of the fuel bed.

H. M. Latham. I think Mr. Bibbins has struck the keynote in regard to bituminous gas producers, when he says that the primary requisites are continuous operation and tar-free gas. There is no question in my mind that these are the most important considerations. Any producer which satisfactorily meets these requirements should have a large field of usefulness.

2 We have already seen from the figures presented by Professor Fernald, that the bituminous producer is at present the predominant type, and it seems probable that future development, especially in large units, will be along this line. In New England the high cost of anthracite coal suitable for use in producers of the strictly anthracite type, offers serious objections to its employment as a fuel.

3 As regards continuity of operation, while it goes without saying that a certain reserve power should be provided, yet it is frequently convenient and desirable in installations where power is required every day in the year, to be able to operate without calling upon the reserve, or in other words, to run absolutely without interruption.

H. H. Suplee. In regard to the question of continuous operation, I think Professor Fernald will remember that we have had a number of gas producers running continuously in this country and elsewhere, not for one year only, but for a number of years, but we did not call them gas producers; we called them blast furnaces. But I hardly think we care to run our producers continuously.

2 In regard to the prevention of clinkers by keeping the contents of the producer in motion, that solution was adopted in the Kitson producer ten or twelve years ago, by means of an inclined grate which was made to revolve slowly. As a result the contents of the producer were kept moving up and down, and at no time did any clinker form. The producer was discontinued, but for other reasons. The inventor of that apparatus based it, he said, on the idea that running water would not freeze, and that in the same way, any substance would be prevented from solidifying by keeping it in continual motion.

3 It must be remembered that in the operation of gas engines, the calorific power of the gas produced is not the essential thing, but rather the value of the charge actually delivered to the cylinder; and this can be made almost anything which may be desired, the proportion of air being regulated according to the richness of the gas so as to give a charge of practically constant heating value.

E. N. Trump. In making tar-free gas all of the valuable by-products are destroyed. If Mr. Bibbins proceeds to burn up the by-products from the gas in the centre of his producer, he will lose from 80 to 90 lb. of sulphate of ammonia per ton of coal, which would pay for a large part of the coal used in his producer, if it were recovered.

2 As to continuous operation: We have had one plant burning from 150 to 155 tons of coal per 24 hours, in continuous operation for the past ten years; the pressure has never been off that plant but once, and then for a period of two hours.

3 If the fuel bed in the producer is agitated, and plenty of steam provided, clinkering is almost entirely prevented. Agitation can be produced by continuously extracting the ashes at the bottom, thus uniformly loosening the bed. Even with a very deep bed almost no poking is required.

4 Our experience has been with Hocking Valley coal, which will not coke. With coking coals it is more difficult to prevent the clinkering, but the agitation by the special mechanism for removing the ashes prevents clinkering to a great extent.

H. F. Smith. While it is of advantage to run continuously, still in most plants it is desirable to start and stop the engines. The majority of manufacturing plants run from eight to ten hours a day, and it is of equal importance to be able to shut the producer down, and to start up again in the morning with a reasonably uniform con-

dition of operation, within thirty minutes, say, of starting the plant. Whether or not the type of producer outlined here is adaptable to meet that condition is open to question.

George D. Conlee. I would like some information regarding the possibility of naphthalene formation by the gas producer. In coke-oven and coal-gas practice, if the heats are sufficiently low to prevent the formation of naphthalene, an excessive production of tar results. Either the one or the other will be present.

2 Regarding the possibility of removing sulphur from gas by reheating, in the manufacture of enriched water gas for illuminating purposes, the gas is passed through checker brick heated to about 1600 deg. fahr. The gas is then scrubbed with water, cooled and passed through iron oxide to remove the hydrogen sulphide. The passage of the gas through the checkers seems to have no effect on the hydrogen sulphide, though it may change some other sulphur compounds to the sulphide.

The Author. In presenting this paper I have had misgivings that it would be considered by some as unduly optimistic. But I hope that I have been absolved from that charge through the simple showing of facts as complete as were at my command.

2 The producer under discussion is more or less the culmination of experiments of many years on different types. It represents the work of a number of engineers who have all striven for the perfection of the bituminous type in one form or another, and I feel safe in saying that the results are such as to give us some encouragement that the problem of gasifying bituminous coal is not as hopeless as it has been supposed to be.

3 First let me define what is meant by continuous operation. While I think no commercial plant should have to shut down every fifth day to clean out, yet 365 days for the plant does not necessarily mean 365 days for the producer. Taking conditions such as normally exist in an electric light plant using steam boilers, we should expect a producer unit to run at least as long without excessive labor charge for cleaning and recharging. A small percentage of reserve equipment is always essential, but 100 per cent is certainly not required.

4 If the producer is to stand by itself, there is no occasion for especial leniency, i. e., we should demand from the designers a grade of service equal to that rendered by present steam plants, and from present indications this can be obtained.

5 These high rates of combustion—30 to 50 lb. per sq. ft. grate area mentioned in the discussion—are interesting, but it must be borne in mind that sometimes the amount of coal fired includes the additional fuel for building new fires. It is apparent from the Norton test that a very considerable proportion of the total coal fed into the producer was withdrawn at the end of a normal run, and if the heat equivalent of this fuel be deducted the rate of combustion will be lowered considerably. So, in comparing intermittent and continuous types of producers, it is necessary to take this extra fuel into account, for in the case of very frequent recharging the net loss is high.

6 The size of producer mentioned by Professor Fernald is rather extraordinary. I think not many of us realize that 3000-h.p. producers are being built. If it was a two-shell producer (the two rated as a simple unit) it should hardly be compared with the single shell producer on the same basis.

7 The sensitiveness of the balanced draft method of control has, I think, been overestimated by Mr. Chapman. While it is stated in the paper that the two control valves should be permanently set, I presume it would be recognized that these valves are put there to correct any inequalities or deficiencies in the fuel bed. When the producer is properly operated the valves need little or no adjustment, otherwise they must be adjusted occasionally.

8 I do not quite agree with Mr. Chapman's statement that it is impossible to maintain uniformity of the fuel beds with hand firing. When the plant illustrated was visited I noted this point especially by the aid of a simple apparatus. This is a double poker, consisting of a section of pipe with a solid rod through the center. By shoving both down into the fire and pulling out the rod and covering the pipe with a glass at the top, the condition could be noted. It was interesting to see that when the top of the fuel bed appeared practically dead, just under the surface it was at the proper temperature. I did not find the irregular conditions of fuel bed which Mr. Chapman mentions and I do not think it was merely a coincidence. The tendency towards short circuiting which he fears in large producers is not as marked as might be expected, excepting with wet peat, possibly owing in part to conditions.

9 As to the sulphide which Professor Rautenstrauch mentions, I can only say that it has not to my knowledge caused trouble. I have seen engines running successfully for a time on by-product coke-oven gas where it was found that by keeping the rods as hot as possible the deposition of sulphur was avoided and the consequent

corrosion of the rods. As far as I know napthalene has not created similar trouble. A napthalene formation is characteristic of the distillate process where the higher hydro-carbons form the greater percentage of the heat value.

10 It is encountered in by-product coke-oven gas to some extent. But the difficulties arising from deposition of napthalene seem to be confined to delicate measuring instruments rather than the engine valves or rods which seem to be at a temperature sufficient to dissipate the accumulation. In producers the heats are run well above the destructive point.

11 Mr. Smith seems uncertain as to the possibility of the producer under description retaining its condition over periods of daily shutdowns. Table 1 shows a period of 18 days—432 hours—during which the producer was entirely idle for 23-hour periods. After a night's shutdown 15 minutes usually suffices to bring the fire into normal temperature conditions.

12 The automatic variation in the proportion of air and gas to the engine according to the richness of gas delivered to it is a problem of engine design relating to regulation of mixture. Designers must face the possibility of variations in gas from the best producers, and I do not believe any mechanical agitation of the fuel bed will avoid this necessity. In a plant employing a 15,000-ft. mixing holder I have observed a puff of rich gas (liberated just after charging) make its way clear through to the engine at regular intervals quite destroying the mixture for the moment.

13 Mr. Trump assumes that the breaking up of hydrocarbons occasions a serious loss of efficiency not encountered in the generators of tar-laden gas. Just what are the precise reactions seems to be unsolved, but in the last analysis only one factor is uppermost—the comparative efficiency of the two systems. I doubt that much over 70 per cent is obtained from either process and less when the power consumption of tar extracting auxiliaries is taken into account.

ECONOMICAL FEATURES OF ELECTRIC MOTOR APPLICATIONS

By Charles Robbins,[1] East Pittsburg, Pa.

Non-Member

The principal object of this paper is to show, by figures and curves, based upon actual tests and investigations of existing installations, how a problem in motor drive can be handled in order to show its maximum economy. It will endeavor to show that the hourly cost of operation is dependent upon the characteristics of the various types of machine tools, from the standpoint of power and time required. The load and time factor of the tool will be taken into account and the influence of this factor on the cost of production. Data are also given upon the electric-motor equipment of machine tools, with suggestions for its standardization.

2 There are certain types of tools in which the operations to be performed require constant speed, for which service the constant-speed type of motor should be used. Other types of tools call for a cycle of duties, in which the range of speed may vary almost from minimum to maximum conditions. In these cases the adjustable-speed type of motor should be used for the greatest economy. There are, therefore, in a single shop two distinct service conditions calling for different types of motors, different methods of applying the motors to the tools, and different methods of control.

3 Where direct current is available, these conditions can be met by the direct-current motor, i.e., both a constant-speed and an adjustable-speed motor is available for machine-tool work. On the other hand, the alternating-current motor is essentially a constant-speed machine. At the present time no commercial method has been found for varying the speed of an alternating-current motor in such a way that it can be

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successfully used for machine-tool service. Thus it is apparent that the machine-tool designer must take into account not only the application of the motor to the tool, but also the class of current supply available in the manufacturing establishment in which the tool is to be used.

4 An alternative method of driving is by the use of a system of gears, commonly called a gear box, driven by a single belt considerably larger than that ordinarily employed on cone pulleys. This large belt will to a considerable extent furnish the power required, the necessary changes in speed being obtained by changing the gears. Obviously, however, a gear box arrangement cannot be as convenient of manipulation as a motor controller, which can be mounted in a position to be easily reached by the operator. In addition to this difference between the two methods of changing speeds, the motor drive offers finer gradations of speed; that is, if the same results were obtained in a gear box, the multiplicity of gears would be considerable, and the up-keep a matter to cause serious consideration.

STANDARDIZATION OF TOOL EQUIPMENTS

5 One of the great drawbacks to a harmonious design of motor and tool has been the lack of a proper understanding of the joint problems of the motor and tool builder, this condition showing the necessity of some standard in respect to speeds or speed ratios, method of control, and certain dimensions of the motor or its adaptation to the tool. There is also a lack of standardization of the method of supplying power in industrial plants. For instance, there are so-called different systems, as direct current of 110, 220 and 500 volts, and alternating current of 220, 440 and 550 volts and two or three phases; also either 25 or 60 cycles may be called for.

6 In view of the above conditions, which are in a measure arbitrary, the future developmentof the art will be materially benefited if some standardization can be adopted by the tool builder and the motor builder, whereby they may be able to recommend certain standard power equipments for metal-working establishments.

STANDARDIZATION TO ACCORD WITH CENTRAL STATION SERVICE

7 Central station power companies now realize the great advantage of a day load and are quoting low power rates to manufacturing establishments. It seems probable that in the future much of the power for small, and to some extent for large, manufacturing estab-

lishments will be furnished by central power companies, either those which are formed for the purpose of furnishing power only, or those which are regularly organized as public utility companies, furnishing both power and light. To this latter type of existing central power stations the day load supported by power service to manufacturing establishments is particularly attractive.

8 As most communities contain both manufacturing plants and central station companies, we look forward to an immense development of central power service, to be used by large manufacturers as well as by the smaller ones. For this reason we suggest that the class of service, i. e., the characteristics of the current supply, should be taken into account when making standards for the operation of metal-working tools.

9 The steam railroad companies as a class have been to a considerable extent the largest single purchasers of machine tools, and it is well to consider the power requirements of such classes of purchasers when deciding upon a standard of motor equipments for tools.

10 For some years it has been the almost universal practice of steam railroad companies to install alternating-current generator equipment in their power stations; these are principally of the turbine type, largely for the reason that their requirements are to a considerable extent similar to those of the central stations of power companies. They are called upon to distribute current for lighting their train sheds, stations and yards, and power for operating turn-tables, transfer tables, etc., and for the operation of their repair shops, which usually consist of machine and wood-working divisions. Because of the simplicity and the great desirability of alternating-current motors, the railroad companies have adopted them almost exclusively for constant-speed service, as exemplified in the machinery of their wood shops, and for miscellaneous power purposes, such as pumping, operation of fans, driving incidental sections of line-shafting for supplying power to the smallest types of tools, on which it would be inadvisable to employ individual motors, and to tools requiring constant-speed motors.

11 For tools whose operation calls for adjustable speed, the standard practice is to employ direct-current adjustable-speed motors, using a controller conveniently located to the operator in such a way that the variation from minimum to maximum speed can be made with great facility, therefore affording a ready means of obtaining the maximum output for which the tool is designed.

12 It will be evident from this practice that two kinds of current are employed—alternating current for the primary and direct current

for all secondary operations. To transform from alternating current to direct current, either a rotary converter or a motor-generator set is employed, the specific selection of one or the other depending somewhat upon local conditions and the class of supervision available for the operation of the outfit.

13 The same scheme of operation can be very advantageously employed when using central station service for the operation of machine shops or metal-working establishments in which machine tools are employed. Such a standardization of tool equipments by the tool builders and the motor manufacturers would tend to place the operation of metal-working tools on a more economic basis, in that it would enable better tool equipments to be designed with a definite certainty that the motor requirements could be forecasted. As it is now, a very considerable risk is involved in designing tools in advance of orders. Few companies manufacture motor-driven tools in large quantities and the public is thus called upon to pay a higher price because of a lack of standardization.

14 On account of the fixed conditions of central station service, it is almost universal for the service to be 60-cycle, 3-phase, and as the transmission line is of relatively high voltage, transformers will be necessary at each industrial plant, and the voltage of the motor installation can thus be suited to the requirements. In metal-working establishments, where the motors can be located on the tools, or to some extent in close proximity to the metal structure, it is desirable to use a relatively low potential, say 220 or 440 volts. Thus in a measure there has been established automatically a standard for alternating-current service, consisting of 60-cycle, 3-phase, 220-440-volt, this standard being that used by most of the largest single purchasers of metal-working tools, i. e., the steam railroad companies.

15 By the adoption of standards which conform with the central station supply service, it is evident that even in the case of very large manufacturers who have their own isolated power plants, use can be made of a so-called break-down connection with the central station power company, as an extra precaution to insure continuity of service. This break-down connection can be made available only when the service supply is uniform with that employed by the isolated plant. Connection to a central power company would prove a very great advantage to a manufacturer for overtime work, when but little power is usually required, or under conditions when but a small percentage of tools are in operation as it would permit closing down the isolated plant, and the operation of the limited service from the outside power system.

16 This standardization of tool equipments would also enable existing manufacturing plants not equipped throughout for electric driving, but requiring the service of machine tools, to make trial installations of motor-driven tools or of a rapid-production tool, in which much of the advantage to be gained is due to the motor drive.

ANALYSIS OF OPERATING CONDITIONS

17 It is only recently that data have been available to show beyond doubt the intermittent operation of the average machine tool. When a machine shop is driven by a belt from engine to lineshaft, and from lineshaft to machine tool, it is difficult to determine with any degree of exactness the length of time any particular tool is in operation, or the average time of operation during the working day.

18 With the installation of motors on lineshafts, it became evident that the total horsepower capacity of motors was much in excess of the power generated in the power station. This ratio is sometimes three to one, other times possibly four to one.

19 As individually driven tools are adopted it is noticed that the total horsepower capacity of all the motors connected to the service grows very rapidly, and that the ratio of the connected capacity to the power supplied is often as high as five or six to one, indicating that the time-load factor of the average machine tool is relatively low.

20 This apparent difference between the connected capacity of motors and the demand on the power station has led to a careful analysis on the part of the motor builders to determine exactly the length of time tools can be expected to be in operation..

21 An analysis which took into account the time of loading, cutting, unloading, and other delays occasioned by miscellaneous causes, showed conclusively that it was not necessary to use a continuously rated motor; in fact, an intermittent rating on the motor for a period not exceeding two hours' continuous service answers for almost all kinds of machine tool applications. This knowledge enabled the motor manufacturer to build a more economical motor, one of smaller size, and consequently reduce the expense of applying motors to machine tools. The present-day tool equipment ought not, therefore, to be much more expensive, if any, than that of the belt-driven tool, when the cost of belting, shafting and power house equipment is considered.

22 When machine tools are equipped with individual motors, a graphic recording meter may be connected in the motor circuit, making it possible to have a complete log of the operation of the particular

tool during its time of service. The chart furnished by the graphic meter will show the time of loading and unloading the tool, the time of cutting, all delays due to stoppage for one cause or another and the amount of power to operate the tool, which is a direct function of the work done.

23 Fig. 1 shows a graphic recording meter by which interesting tests have been made in studying machine tool operations. The instrument is unlike an indicating meter, in that instead of a needle

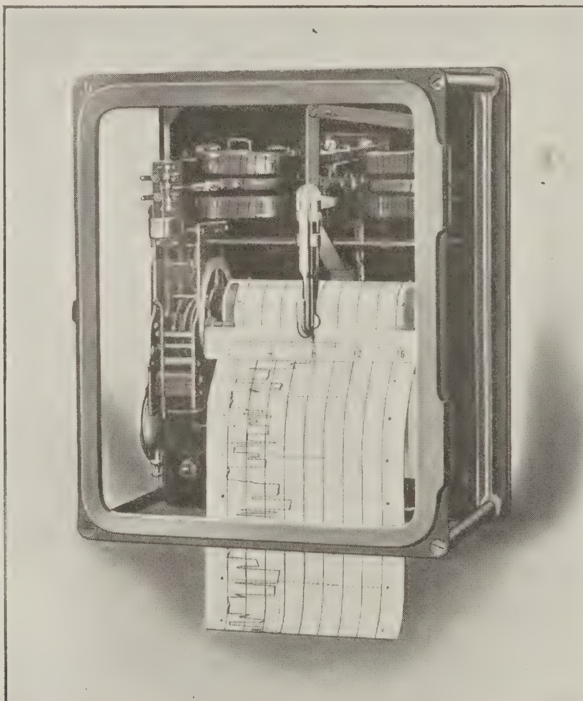


Fig. 1 Graphic Recording Meter

passing over an indicating scale, the meter is provided with a pen moving horizontally, thus making a line on a properly graduated roll of paper. The paper is moved by clockwork, vertically, and at right angles to the pen, so that a permanent record of the magnitude and time of all operating changes is obtained.

24 A time study can be made from each tool from these charts, and knowing the theoretical time for the job an analysis can be made of the

curve, furnishing information that will enable the foreman to increase the productive capacity by the elimination of delays. He will also know whether or not the tool has been working at its maximum capacity, whether the tools have been kept up to standard conditions, and in general can apply the necessary remedies.

The Economics of Motor Drive as Determined by the Actual Performance of the Tools

25 The economy of the individual motor drive, due to the fact that practically the exact cutting speed can be obtained for any operation, has been pointed out. This economy is not so important, however, as that of keeping a tool in continuous operation through longer periods of time, by reducing the time required for handling and other avoidable delays, as previously mentioned.

26 The accepted method of capitalizing motor drives seems in general to be on the basis of the incidental savings in the workman's time. In our opinion this is not the whole story by any means. When determining the monetary advantage of motor drive, the value of time saving should be considered on the basis of its effect on the total cost, which includes the workman's labor and the investment cost per hour of the tool.

27 In addition to workmen's wages, every shop has the following expenses:

- *a* Interest and depreciation on cost of buildings and accessories.
- *b* Repairs and renewals to existing equipment.
- *c* General operating expenses, including losses due to defective workmanship, design and material.
- *d* Salaries of supervisors, engineering staff and clerks.

28 These overhead charges must be included in the cost of any manufactured article. A method frequently employed is to determine from time to time the percentage which the total overhead charge bears to the cost of total actual or productive labor. This percentage in large shops reaches from 100 to 200 per cent, or even more. The total labor charge is then obtained by multiplying the actual labor cost by one, plus the per cent to be added for the overhead charge.

29 This is an easy way to take care of the overhead charge; but the method is inaccurate and does not show the relative importance of different types and sizes of machines. This statement is especially true where a great variety of materials is manufactured, in shops using a

large number of different types and sizes of tools. Under such conditions, the precentage obviously varies within wide limits for different kinds of work.

30 A satisfactory method of distribution is to set off against each tool its proportion of the total overhead charges. The portion chargeable to each tool depends entirely on local conditions; and thorough amiliarity with these conditions is needed in order to apportion these charges equitably. In this way, the relative importance of each machine is taken care of.

31 In a shop where only one type of article is manufactured, and the castings are passed from one machine directly to the next, a simple and logical way is to divide the total overhead charge among the tools, in proportion to the floor space charged to each tool. In the majority of shops, however, the above simple condition does not exist; several sizes and kinds of articles are usually turned out, and various sizes and types of tools, differing greatly in their operating characteristics, are employed. In such cases, not only must the floor space be considered, but also the time each tool is actually in operation, the nature of the work and the amount of supervision and engineering attention needed.

32 Large shops handling different classes of materials are in most cases divided into various departments or sections, and each section may be considered as a separate smaller factory. The overhead charges against each department may thus be apportioned among its tools in proportion to the floor space occupied, making proper allowance for special local conditions, or special supervision or engineering attention. Here again is required thorough familiarity with both the engineering and the shop features of the materials manufactured.

33 In our experience we have found the overhead charges to be approximately as follows:

Variable charges	from 50 to 55%
Salaries	from 25 to 30%
Interest on cost of machine tools	from 5 to 10%
Depreciation on cost of machine tools	from 5 to 10%
Fixed charges	3%
Power	from 1 to 2%

The detail method of arriving at these general figures is found in Appendix 5.

DEFINITIONS OF TERMS

34 In discussing the economics of motor drive there will be a number of terms used which are here given with our interpretation of their meaning.

Applied to the operation:

Time factor = ratio of actual cutting time to total time required to complete a machining operation

$$= \frac{\text{Actual cutting time}}{\text{Total time to complete operation}}$$

Applied to a Machine Tool:

$$\left.\begin{array}{l}\text{Time factor}\\ \text{in per cent}\end{array}\right\} = \frac{\text{Total daily actual cutting time in hours}}{\text{Total number of working hours}} \times 100$$

Average running load = average input to motor while operating, usually expressed in kilowatts, but may be expressed in percent of full load input.

For rough calculations in this paper the input of a motor in kilowatts is assumed to be the same as the output in horsepower; that is, the motor efficiency in all cases is assumed to be about 75 per cent. This low percentage will take care of the fact that motors operate at light loads a considerable part of the time.

Maximum load = maximum input to motor, expressed in same terms as the average running load.

Average load = *Average daily load* = average input to motor during the total working hours; usually expressed in kilowatts. This load multiplied by the total number of working hours gives the total kilowatt hours consumed per day, and is the basis of payment for energy. The average load multiplied by the number of hours per day and by the price per kilowatt-hour gives the cost of energy per day. The average load also equals the average running load multiplied by the time factor.

Load factor = the ratio in percent of the average daily load to full load rating of the motor, or

$$\text{Load factor} = \frac{\text{Average daily load}}{\text{Full load rating of motor.}}$$

CONDITIONS ENTERING INTO THE OPERATION OF A MACHINE TOOL

35 In order to obtain a maximum output from a machine tool, a careful analysis must be made of all the conditions entering into the operation of the tool. One method of doing this in the case of a motor-driven tool is to take power readings at frequent intervals and lay

these out on a chart basis. Another and a much more convenient method is the employment of a suitable meter, as already described, designed to make a graphic curve, showing the exact condition occurring in the service when such a meter is applied to any motor-driven tool.

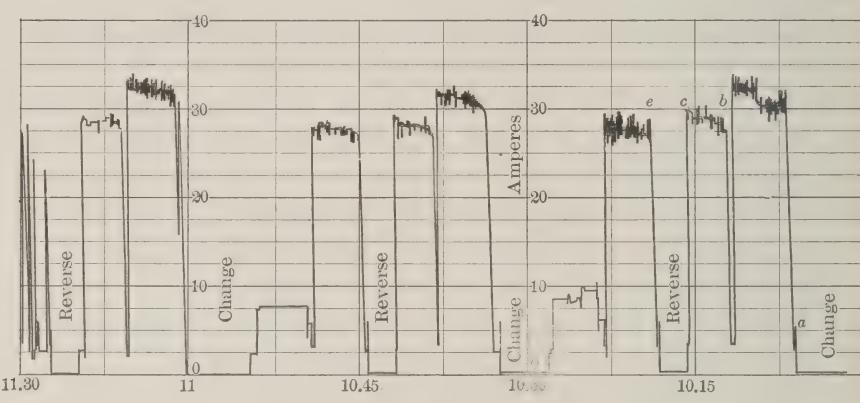


FIG. 2 METER RECORD WHEN TURNING SHAFTS SHOWN IN FIG. 3

36 Fig. 2 shows a record obtained while shafts of the dimensions shown in Fig. 3 were turned from machinery steel. Both Fig. 2 and Fig. 3 are lettered for reference. The records read from right to left, as indicated by the time at the bottom of the curve in Fig. 2. The vertical coördinate is in amperes, the full scale being 50 amperes. This cur-

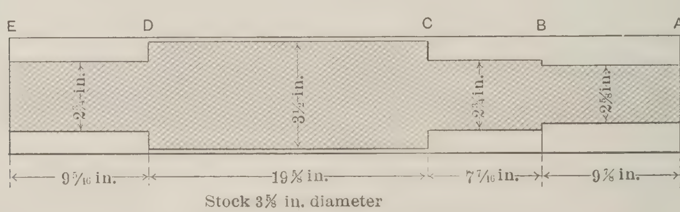


FIG. 3 SHAFT OF MACHINERY STEEL

rent at 220 volts corresponds to 11-kw. input to the motor. At the extreme right, the record indicates zero power; that is, the motor was standing idle.

37 During the interval marked "change," the stock to be turned was placed in the chuck of the lathe. At *a* the current increases for a

very short interval to about 3 amperes, while the lathe was running idle. The current then suddenly increases to about 30 amperes, due to the fact that the cutting tool was fed against the stock and the cut started. The current remains at this value for a period of about five minutes while the cut *AB* is taken, changing the diameter of the stock as indicated in Fig. 3. At *b* the current drops to three amperes, the motor running idle while adjustments of cutting tools are made. The current then increases to 28 amperes while the cut *BC* is taken. At *c* the machine is stopped to reverse the half-completed shaft for machining the opposite end. At *e* the machine is again started and the current increases to 27.5 amperes while the cut *ED* is taken. Another adjustment of the diameter is then made, the machine running idle for a short interval. From 8 to 10 amperes are required when the final cut *DC* is taken, after which the machine is stopped to remove the completed shaft. A similar cycle is repeated when the next shaft if turned.

38 The record shows three completed cycles, covering the time required to complete three shafts At 11.15 a.m., before taking the cut *ED*, there are sudden fluctuations of current; the form of the curve compared with other cycles shows clearly that some trouble was encountered with the cutting tool or work, and the adjustments made. The record also shows the delay in time.

TABLE I. ANALYSIS OF TIME AND POWER OF A LATHE OPERATION

Shaft	Time	Mins. Amps.	Cutting				Mins. %	Total Cutting	Change	Reverse	Adj. Tool	Misc.	Comp.	Time Factor	Load Fact.
			AB	*BC*	*ED*	*DC*									
1	7.30	Mins.	5.1	3.7	4.9	4.9	Mins.	18.6	5	5	12.0	21.2	61.8		
		Amps.	23	22	22	5	%	30	8.1	8.1	19.5			30	12
2	8.05	Mins.	5.3	3.9	4.4	4.4	Mins.	18.0	4.7	2.4	4.4		29.5		26
		Amps.	25	23	24	5	%	61	15.9	8.1	14.9			61	
3	8.30	Mins.	5.0	3.7	4.8	4.6	Mins.	18.1	7.5	2.4	1.9		29.9		
		Amps.	29	25	24	7	%	60.5	25	8	6.4			50.5	26.5
4	9.05	Mins.	4.5	3.4	4.8	4.7	Mins.	17.4	3.2	8.9	2.3	27.1	31.8		
		Amps.	31	29.5	24	6	%	54.8	10	27	7.2			54.8	25
5	10.05	Mins.	5.1	3.8	4.5	4.7	Mins.	18.1	5	2.5	1.3		54.0		
		Amps.	29	26	25	6	%	33.5	9.3	4.7	2.4			33.5	14.6
6	10.30	Mins.	4.9	3.6	4.6	4.9	Mins.	18.0	4.3	2.4	2.0		26.7		
		Amps.	28	25	25	5	%	67.5	16.1	9	7.5			67.5	25
7	11.00	Mins.	5.0	3.7	5.4	5.1	Mins.	19.2	5.5	2.6	2.7		30.0		
		Amps.	29	26	24	4.5	%	64	18.4	8.7	9			64	27

39 Table 1 is a summary of the data obtained from the graphic record, part of which is shown in Fig. 2. Observations of cutting

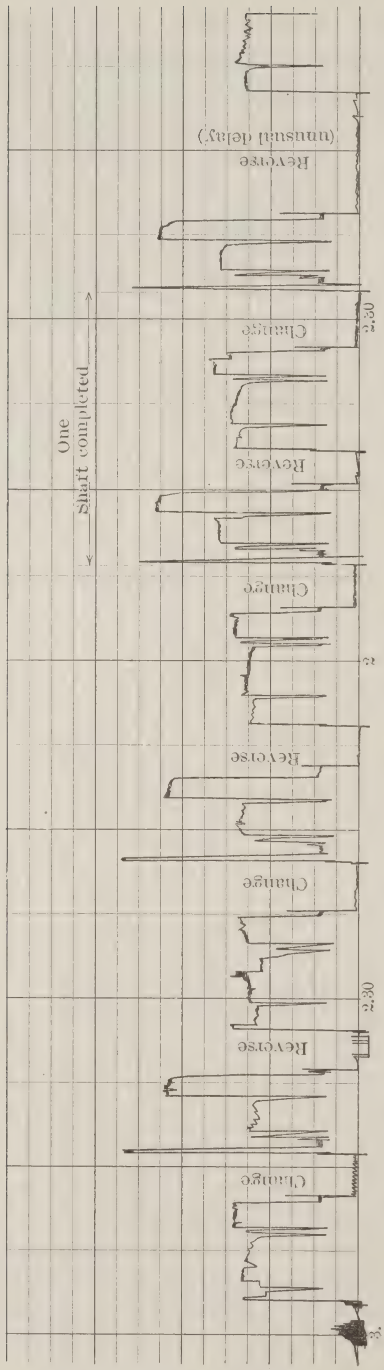


FIG. 4 RECORD MADE WHEN TURNING A LIGHT SHAFT WHICH COULD BE PLACED AND REMOVED BY HAND.

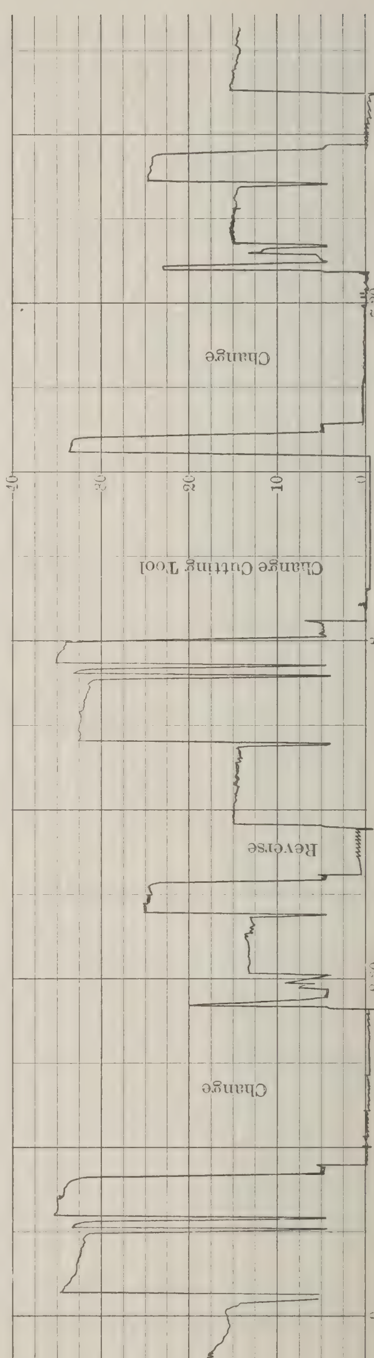


FIG. 5 RECORD MADE WHEN TURNING A HEAVY SHAFT WHICH REQUIRED A CRANE FOR HANDLING

speed and feed were taken at the lathe. The cutting speed used while turning these shafts was 55 to 60 ft. per min. The feed while taking the cuts *AB*, *BC*, and *ED* was 0.04 in. per revolution, and while taking cut *DC* was 0.077 in. per revolution. The normal time to complete a shaft was from 27 to 32 min. In case of shaft No. 1 the time was 62 min.; this was the first shaft turned after starting work, and preliminary adjustments, oiling lathe, etc., consumed 21 min.; 12 min. were required to adjust the cutting tool. In the case of shaft No. 5, 54 min. were required on account of a 27-min. delay. The amperes referred to in Table 1 are those below the 3 amperes required to run the machine idle; they are, therefore, a measure of the power required to remove the metal. The time factor averages 53 per cent; its maximum value is 67 per cent, and its minimum value is 30 per cent. The load factor is 25 per cent under normal conditions.

40 It must be obvious that, with a given rate of cutting, the fewer the delays, the higher will be the time factor. The magnitude of the records is an indication of the rate of removing metal, as will be further explained.

41 By means of this meter record it is possible to discover all delays, and to check the rate of cutting metal. Those are the two fundamental factors which determine the rate of output on machine tools. Any deviation from the standard cycle of operation is at once detected from the form of the record. Observations of cutting speed and feed need be taken in only one case. The record will not only show the deviation therefrom, but will also indicate whether the modification is an improvement or a drawback to the rate of output. Fig. 4 and Fig. 5 show two records taken on the same roughing lathe, operated by the same man, but turning two different shafts. The shaft turned while making the record shown in Fig. 4 was light, and could be removed and replaced in the lathe by hand. That turned when the curve in Fig. 5 was obtained was heavy, and required crane service. The greater intervals between cutting operations, so apparent in the case of Fig. 5, were due to delays in obtaining crane service to handle the heavy shaft.

42 Records taken during several days of operation showed an average of 5 min. longer time required for every change and reversal when made by crane. This condition was remedied soon after making the test, by installing a jib crane next to the lathe, and the time to complete the larger shaft was thereby reduced from 55 min. to 45 min., a saving in time and cost of about 20 per cent. The overhead charge against this tool (see Appendix 5) was 60 cents per hr., or $6 per 10-hr. day. The operator received $3.50 per day, making a total daily

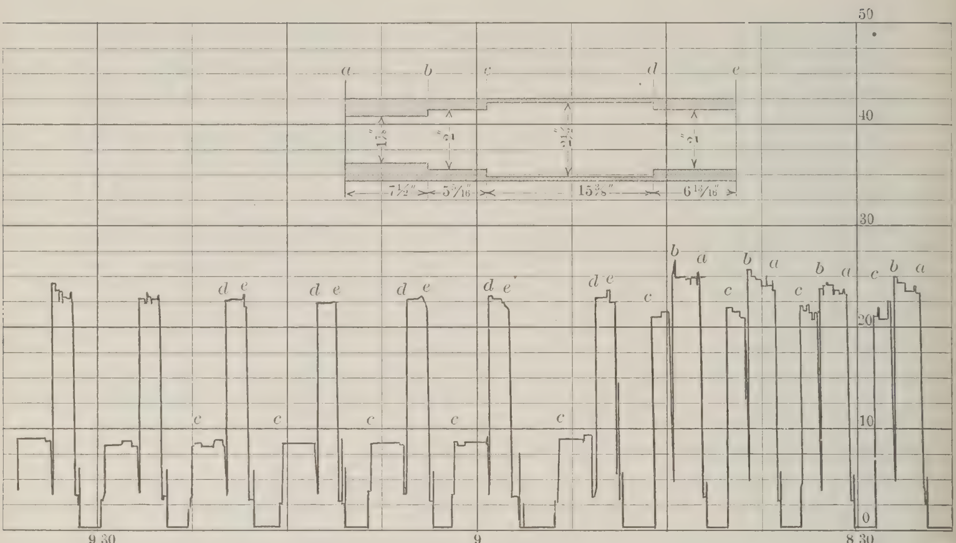


Fig. 6 Record when Turning the Shaft Shown in this Diagram

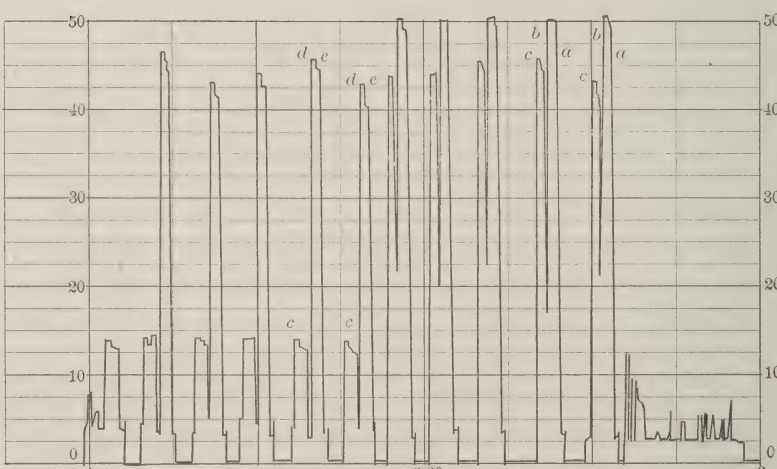


Fig. 7 Record when Turning the Same Shaft at Double the Cutting Speed

expense of \$9.50. Before the jib crane was installed ten shafts were completed per day, making the cost of actual labor and overhead tool charge \$9.50 divided by 10, or 95 cents per shaft. After the improvement, 12 shafts per day were completed with the same overhead charge, thus reducing the labor and overhead tool charge to 79 cents per shaft.

43 Such a delay seems self-evident, after it has been discovered, but in a large shop where everybody is busy small delays are ea ily overlooked. An automatic recording meter reveals delays caused by grinding and replacing tools, etc., besides those just indicated.

44 The elimination of delay, however, is not the only advantage to be obtained from the use of a recording meter. Fig. 6 and Fig. 7 are meter records which show rates of cutting on a shaft with the dimensions given in Fig. 6. In Fig. 6 the cutting speed was 50 ft. per min. The feed for cuts *AB*, *BC* and *ED*, was 0.05-in. per revolution, and for *DC* was 0.072 in. per revolution. The same feed was employed for corresponding cuts in Fig. 7, but the cutting speed was 100 ft. per min. It will be noted that the current above friction load in Fig. 7 is double that required for similar operations in Fig. 6. The saving in time is clearly shown. An analysis of records of this kind, taken over a period of several days, gives a means of determining the most economical feeds and cutting speeds to employ on a given operation.

TABLE 2. TIME FOR ROUGHING SHAFT, EXTREME CONDITIONS

	Average Condition		Best Conditions		Poorest Conditions	
	Minutes	Per Cent of Total Time	Minutes	Per Cent of Total Time	Minutes	Per Cent of Total Time
Removing and replacing shafts	6.0	38	2.8	23	14.9	50
Adjusting tool	1.7	11.0	1.4	12	6.7	22
Cutting	8.0	51.0	8.0	65	8.4	28
Total	15.7	....	12.2	....	30.0	....

45 Table 2 shows the time relation between the various operations in roughing the shaft, outlined in Fig. 6. Approximately the same conditions were found with shafts of other characteristics. The time factor of lathe operation for this class of work is thereby shown to vary from 25 to 65 per cent, the average being about 50 per cent.

46 An investigation by personal observations over a short period of time often leads to erroneous results, as is shown by the following

experiments: In turning shafts on a roughing lathe, the first trial was with a cutting speed of 80 to 100 ft. per min., and a feed of 0.026 to 0.044 in. per revolution. In the second case, a cutting speed of 40 to 50 ft. per min. at a feed of 0.05 to 0.07 in. per revolution was employed. A single job could be completed in either case in 16 min., 12 min. being required for cutting. However, the average time per shaft, during several days operation, was 22.6 min., with the higher speed, and 21.6 min. with the slower speed. The same number of cubic inches of metal per minute was removed in each case; but with the higher speed, more frequent regrinding of tools was necessary, resulting in more delays and giving the lower speed five per cent advantage in time saving.

SUMMARY OF THE USES OF THE GRAPHIC RECORDING METER

47 By means of the graphic recording meter, the following improvements in shop management can be effected:

a If individual motors are used to drive machine tools, the exact percentage of total working hours consumed in actual cutting can be determined; it is found to average from 40 per cent to 50 per cent, the maximum being as high as from 60 per cent to 65 per cent where the cut is of long duration; the minimum from 20 per cent to 30 per cent where jobs are short and the delay long in waiting for material, drawings, etc.

b The meter reveals all delays and suggests measures for eliminating those not essential and reducing all others to the minimum, thus materially increasing the time factor. All delays shown should be accounted for, and an attempt made to avoid them. Common delays are in assignment of the next job, in obtaining drawings, tools and other necessary materials, and in waiting for crane service.

c The rate of cutting indicated by the power consumption of motor-driven tools can be checked with a recording meter. The maximum rate is limited only by the nature of the work, the strength of the machine tool and of the cutting tool.

d The rate for maximum economy can be determined for different classes of work; and the records, considered as standard, can be compared with other operations of the same

character to see whether the proper rates of cutting were used. In a finishing operation the rate depends upon the accuracy required. A record can be made while an expert machinist does the job, and this record should be referred to when other jobs of similar character are machined.

48 By the use of curve-drawing meters, and a careful study of the data obtained, the superintendent of a shop in which the individual motor-driven system is employed can set a limit fair both to employer and employees, for roughing, finishing, adjusting and setting-up. Different methods of doing the same job can be compared to determine which is the most efficient.

49 The graphic meter need not be located near the machine to which it is connected, but may be placed in the foreman's office. Small leads connected to a shunt, or to a series transformer, according to whether direct current or alternating current is employed, are all the wiring required. The wiring can be so arranged that the connections of the meter can be readily transferred to any one of several tools; thus a single meter can be made to serve a group, or any number of tools, depending somewhat upon the frequency with which the records are required.

50 So far we have dealt chiefly with the time required to do machining operation, time being a most important consideration with shop managers and those who use machine tools. The power consumption, however, is also of some importance, especially to those requiring motors for machine tool operation.

RELATIVE ECONOMY OF LINESHAFT DRIVE AND INDIVIDUAL MOTOR DRIVE

51 An increase in economy of operation of manufacturing machinery can be effected in two ways: first, by reducing the power required to operate the machinery, by saving of friction load, etc.; second, by reducing the time required for a given operation, or, in other words, increasing the output in a given time. When confronted with the problem of deciding between the continued use of an existing lineshaft drive, or an individual motor drive, or when deciding between the two methods for a new installation, the problem should be considered in all its phases, as outlined in Table 3. This table includes every important item to be considered, except one; and in every case the advantage is with the motor.

52 Comparing the first cost is possibly the first consideration to enter the mind of most men, and this is the one consideration omitted from Table 3. That this consideration is relatively of minor

TABLE 3. COMPARISON OF LINESHAFT DRIVE AND INDIVIDUAL MOTOR DRIVE FOR MACHINE TOOLS

Item	Lineshaft Drive	Individual Motor Drive	Advantage of Individual Motor
1 Power consumption....	Constant friction loss in shafts, belts and motor, power for cutting	Friction loss (motor and tool only); useful power only while working	Less power required
2 Speed control...........	No. speeds = no. cone pulleys × no. gear ratios	No. speeds = no. controller points × no. gear ratios	More speeds possible; time saved in speed adjustments
3 Reversing.............	Clutch and crossed belt	Reversible controller	Time saved in reversing
4 Adjusting tool and work	Stopping at any definite point, very difficult	Can be started in either direction and stopped promptly at any point	Time saved in setting up and lining up
5 Speed adjustment......	Large speed-increments between pulley steps	Small speed-increments between controller steps	Time saved by obtaining proper cutting speed
6 Size of cut.............	Limited by slipping belt; large belts hard to shift	Limited by strength of tool and size of motor	Time saved by taking heavier cuts
7 Time to complete a job..	....	....	Much less time required as indicated for previous items
8 Liability to accidents...	Slipping or breaking belts; injury to machine tool; cutting tool or prime mover	Injury to machine tool, cutting tool or motor	Much less liability to accidents
9 Checking economy of operations	Close supervision required; very difficult to locate causes of delay	Accurate tests possible by means of graphic meter which records automatically all delays and rate of cutting	Causes of delay and remedies easily located without personal supervision
10 Flexibility of location..	Location determined by shafting, and changes difficult	Location determined by sequence of operations; changes readily made	Greater convenience in handling and increased economy of operation; more compact arrangement possible

importance will be evident, when the saving in power consumption, and in time, made possible by individual motors, has been considered.

SELECTION OF MOTOR AND TOOL EQUIPMENT

53 In the selection of a motor-driven tool, there are certain features which should be taken into account and properly analyzed, and specifications drawn to cover them. If a tool is for specialized manufacturing, there should be specified:

a The exact class of work which the tool is to accomplish.

b If the power required to remove the metal is not known, then a statement should be made as to the approximate feed and cutting speeds to be taken.

c Careful analysis should be made of the time required to load and unload the machine, to determine the feasibility of employing auxiliary means other than manual labor for loading the tools.

d From this information, an approximate determination can be made as to the intermittency of operation of the tool, in order to decide whether an intermittently rated motor or a continuously rated motor will be required.

e By a knowledge of the physical shape of the work, determination can be made as to whether an adjustable-speed motor will result in economy of time, if used on this particular class of tool.

f Will enable the tool builder to determine upon the proper type of controller, and its most desirable location from an operating point of view for the workman.

54 If a special type of tool is not desired and it is preferable to purchase one with such characteristics that it can be used for general manufacturing, one should determine as nearly as possible the range of material or work for which it will be used in straight manufacturing operations. A knowledge of this will undoubtedly permit of a better motor and tool selection, than the simple purchase of a standard stock tool.

55 It should be realized that under present schemes of operation few tools are in operation more than 50 to 60 per cent of the time, whereas, the load factor of those tools may be as low as from 10 to 40 per cent. Thus we have it brought home to us clearly that much of the time the tool is in idleness and is often operated much less than its maximum capacity.

56 The direct-current motors are built for speed adjustment over a range of 1 to 2, 1 to 3, and in some instances 1 to 4. With the proper selection of controller the speed adjustments may be made in small increments of from 10 to 15 per cent, and since these small increments of speed adjustment are available, it is essential that a controller be selected of such type that it can be mounted conveniently to the operator, so that he may take full advantage of them.

57 Where it is necessary to employ the alternating-current motor, it may be absolutely essential to employ a gear box to obtain the various speed adjustments. When such a machine is employed, the fine gradation of speed obtainable with a direct-current adjustable speed motor is absent, and the gear box will practically take the place of the ordinary cone pulley arrangement. It has, however, one advantage when motor-driven, and that is, that the tool is supplied with positive power at a l times, and will take care of the maximum conditions without slipping or loss of power, which frequently occurs when belt drive is used. In some instances it has been found possible to make good use of the so-ca led multi-speed alternating-current motor. This form of motor consists in certain different types of windings, permitting of a multiple method of pole grouping, such as for instance, a speed of 1800, 1200, 900 and 720 r.p.m., according to the method of winding the motor. In some cases, this type of constant-speed motor, when used in conjunction with a gear box, will permit of somewhat finer gradations of speed than are possible with a constant-speed alternating-current motor and a standard gear box.

58 While it is apparent that with the constant-speed motor all of the advantages of the adjustable-speed motor cannot be obtained, a tool equipped with either type has the advantage to be derived from the ability to obtain a graphic log of the time of operation of the tool. With either type, in combination with a graphic recording meter, a distinct gain can be made over a belt-driven tool from which such graphic curves cannot be conveniently obtained.

59 In Appendix 5, there are the segregated charges, which must in some manner be taken into account in determining the cost of a machine tool hour, not only including the workman's time, but also the actual expense to a manufacturing establishment of having a tool equipped and ready to be used when the workman requires the services of such tool. Table 2 of this appendix will show the range of the tool-hour rate, from which it is evident that it is far in excess of the labor rate for that tool; consequently, any time which can be saved in keeping the tool in its maximum productive capacity will far outweigh any

saving that can be made in the actual labor account. It is this one feature in which the motor-driven tool in combination with the graphic recording meter is destined largely to decrease the cost of machining operations when the records available by this combination are carefully studied and proper remedies applied.

GENERAL CONCLUSIONS

60 The economical operation of a machine shop requires a thorough analysis of all the operating costs; that is, overhead and operating charges of all kinds, and an accurate knowledge of the operating conditions of all machine tools. Investigations of these conditions must be conducted by someone familiar with both the engineering and the shop features of the apparatus manufactured. The investigator should also be familiar with the characteristics of the various types of motors and methods of control, in order that the most advantageous electrical equipment as well as the best machine tool equipment may be installed, with suitable tools for different sets of conditions.

61 Such investigations lead to the following improvements which result in increased productive capacity:

- *a* More flexible arrangement of tools.
- *b* Greater facilities for handling materials at the tools.
- *c* Greater facilities for handling materials between tools.
- *d* Better facilities for obtaining auxiliary material, drawings, tools, etc.
- *e* Better facilities for making adjustment of the tools during machinery operation.
- *f* Removal of causes of unsuspected or avoidable delays due to small accidents and improper characteristics of the drive.
- *g* All lost time, due to whatever cause, and which can be avoided, is immediately brought to the attention of the superintendent, and an analysis of these losses will result in their elimination.

62 With motor-driven tools this analysis can be made much more conveniently and with less expense than can similar studies with any other form of machine-tool drive.

63 While in many shops there are elaborate systems of time keeping, with time clocks, etc., all of which are based on keeping an exact record of the workman's time and seeing that he works the maximum

or full shop time, yet the most important consideration in manufacturing with machine tools is that the tools shall operate their full capacity, on account of their greater hourly value.

64 In comment on this conclusion it may be said that our tests have not been confined to metal-working tools, etc. We have found similar conditions in the wood-working industries, to some extent in cement mills, steel mills, brass and copper rolling mills, to a less extent in the textile mills, where it is a supposition that every machine is running the maximum number of hours, and at its maximum load at all times; and in several minor industries, in which the information therein contained is no less important, even though it might be different than that obtained with machine tools or metal-working tools, as ordinarily installed.

65 Certain it is, that a careful analysis and study of conditions which are conveniently possible in motor-driven establishments will greatly reduce the cost of operation, and it seems reasonable to suppose that the methods herein illustrated may serve some useful purpose if the data will arouse an interest on behalf of those present.

66 We know that wherever the tests have been made, that the conditions of operation have been very materially benefited, and feel without question of a doubt that many dollars ha e been saved on account of the knowledge shown by a simple record taken from motor-driven machines, which records are available to all those who have these meters.

67 The writer wishes to acknowledge his indebtedness to Mr. A. G. Popcke, who has made the tests herein illustrated, and who has supplied some of the information contained in the paper, and without whose coöperation the information herein submitted would not be available.

APPENDICES

The five appendices which supplement the paper pertain to the following subjects:

(1) The characteristics of various machine tools as shown by diagram from recording meters;

(2) Data on the power required to remove metal under the conditions set forth in the appendix, together with convenient charts for determining the various factors mentioned;

(3) A summary of the average horse power equipment for different types of tools and the approximate speeds of the motors, which

are normally selected for this work, the figures given are for average conditions only and are not applicable to the heaviest types of tools. In some instances, also, the motors called for are larger than would be used for tools of several years ago. The object of the figures is simply to indicate approximately the sizes of motors usually specified;

(4) Calculation to determine whether it is more economical to equip an old machine with a motor or to purchase a complete new motor-driven equipment.

(5) Over-head charges and machine hour rates.

APPENDIX NO. 1. POWER ANALYSIS OF MACHINE TOOLS

The results which follow were obtained from graphic meter records from certain machine tools under the conditions specified. These examples are given to show the characteristics of the power and the time factors that enter into the performance of machine tools of different types. (In connection with this Appendix see definitions of terms in Par. 34 of the paper.)

VERTICAL BORING MILLS

2 In Fig. 2 which follows is a record from a 72-in. boring mill equipped with a 220-volt adjustable-speed motor, 780 to 1560 r.p.m., 8.5 h.p. (assumed input at full load 8.5 kw.), taken while the tops of bronze discs were turned off and bored out. The vertical lines at frequent intervals show where the motor was started and immediately stopped, in order to make adjustments

TABLE 1 OPERATING CONDITIONS OF 72 IN. VERTICAL BORING MILL

Test No.	Time Factor %	Average Running Load		Max. Load		Avg. Kw. per Hr.	Load Factor
		Kw.	% Full load	Kw.	% Full load		
1	35	1.8	21	7.7	90	0.63	7.5
2	56	2.0	24	8.8	103	1.1	13.5
3	62	2.8	33	8.8	103	1.7	20.5
4	46	2.8	33	6.6	78	1.3	15.
5	28	2.2	26	7.7	90	0.62	7.5
6	37	2.2	26	8.3	98	0.81	9.5
Avg........	44	2.3	27	8.0	94	1.00	12.

by moving the table of the boring mill a short distance. The gradual decrease in power after 9 a.m., and also after 3 p.m., shows improper use of the controller. The tool was fed towards the center of the mill, thereby gradually decreasing the diameter of the work. The motor was evidently allowed to run at a constant speed, while the speed should have been gradually increased to compensate for decreased diameter of work, and thereby keep the cutting speed constant. The controller was arranged to give the required speed adjustment, and failure to take advantage of this feature caused loss of time. In one

instance the records showed that 10 min. was consumed for an operation which could have been performed in 6 min., if the cutting speed had been kept constant, a saving of 40% in this operation. In another case, the time taken was 15 min., and would have been 9.5 min. with a constant cutting speed, which would have resulted in a saving of 37%.

3 While taking roughing and finishing cuts on this boring mill, from castings of motor frames, brackets and end plates, the conditions were found to be as in Table 1, from which the following summary is obtained:

Average time factor..44%
Average running load ....2.7 kw., or 27% of full load
Maximum load, sustained for several minutes . .8 kw., or 94% of full load
Maximum load peak ....8.75 kw., or 103% of full load
Average load...............................1 kw., load factor 12%

CHARACTERISTICS OF RADIAL DRILLS

DRILLING AND COUNTERBORING

4 Table 2 was made up from records taken upon a 10-ft. radial drill driven by an induction motor of $7\frac{1}{2}$ h.p. 720 r.p.m., obtained when cast-steel pole shoes were counterbored, as indicated in Fig. 1.

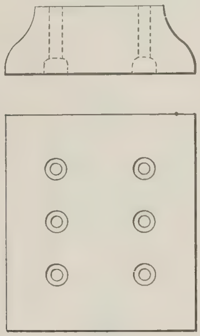
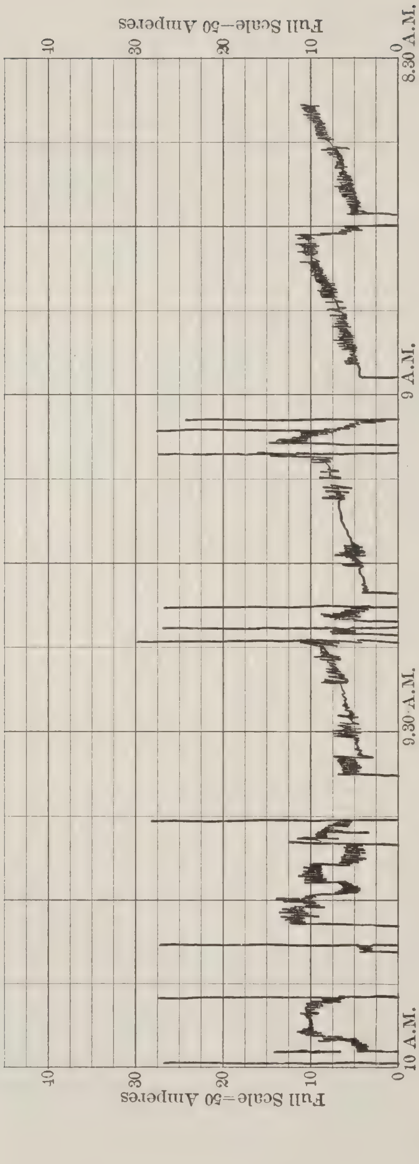
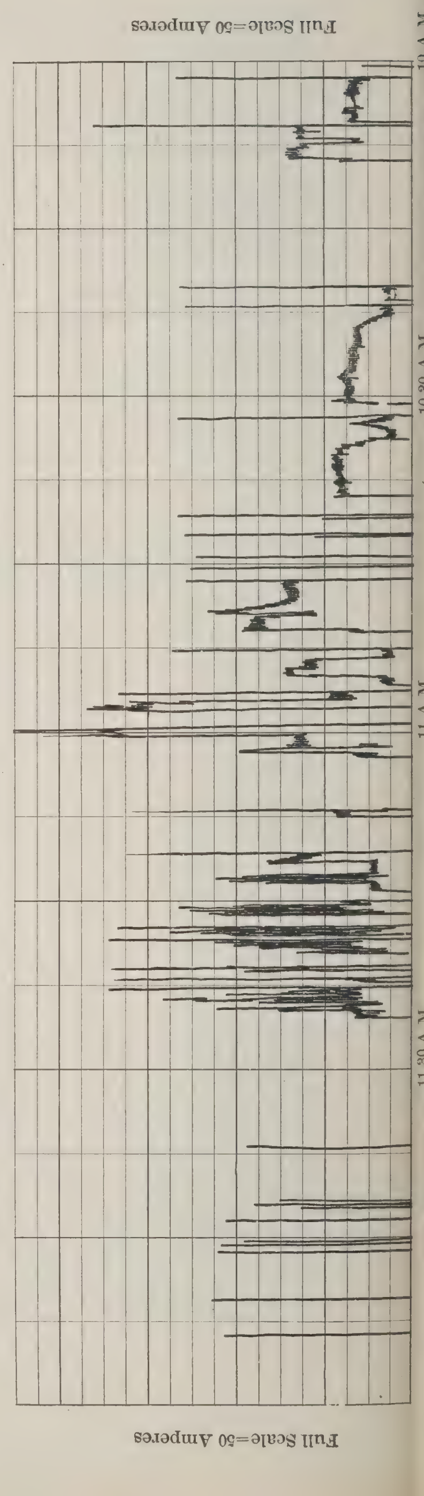


FIG. 1 POLE SHOE, DRILLED AND COUNTER BORED

5 The record (not here printed) shows $7\frac{1}{2}$ min. required to counterbore each of the six holes per pole shoe. The sum of these is about 44 min. The total time for adjusting the drill is given under column headed "Adjust," and was from 15 to 24 min. for each pole shoe. The time consumed in removing and replacing the shoes in the clamps is tabulated under column headed "Change." This ranged from $7\frac{1}{2}$ min. to 10 min. The time to complete counterboring each pole shoe varied from 70 to 75 min. The actual cutting or drilling time was from 57 to 64% of the total time to complete a pole shoe. From 10 to 14% of the time was consumed in changing the shoes, and from 22 to 33% was consumed in adjusting the drill. The average power consumed was 2.8 kw., making a load factor of about 37%.


Full Scale—50 Amperes
8.30 A.M.
9 A.M.
9.30 A.M.
10 A.M.
Full Scale=50 Amperes
0
10
20
30
40


Full Scale=50 Amperes
10 A.M.
10.30 A.M.
11 A.M.
11.30 A.M.
Full Scale=50 Amperes

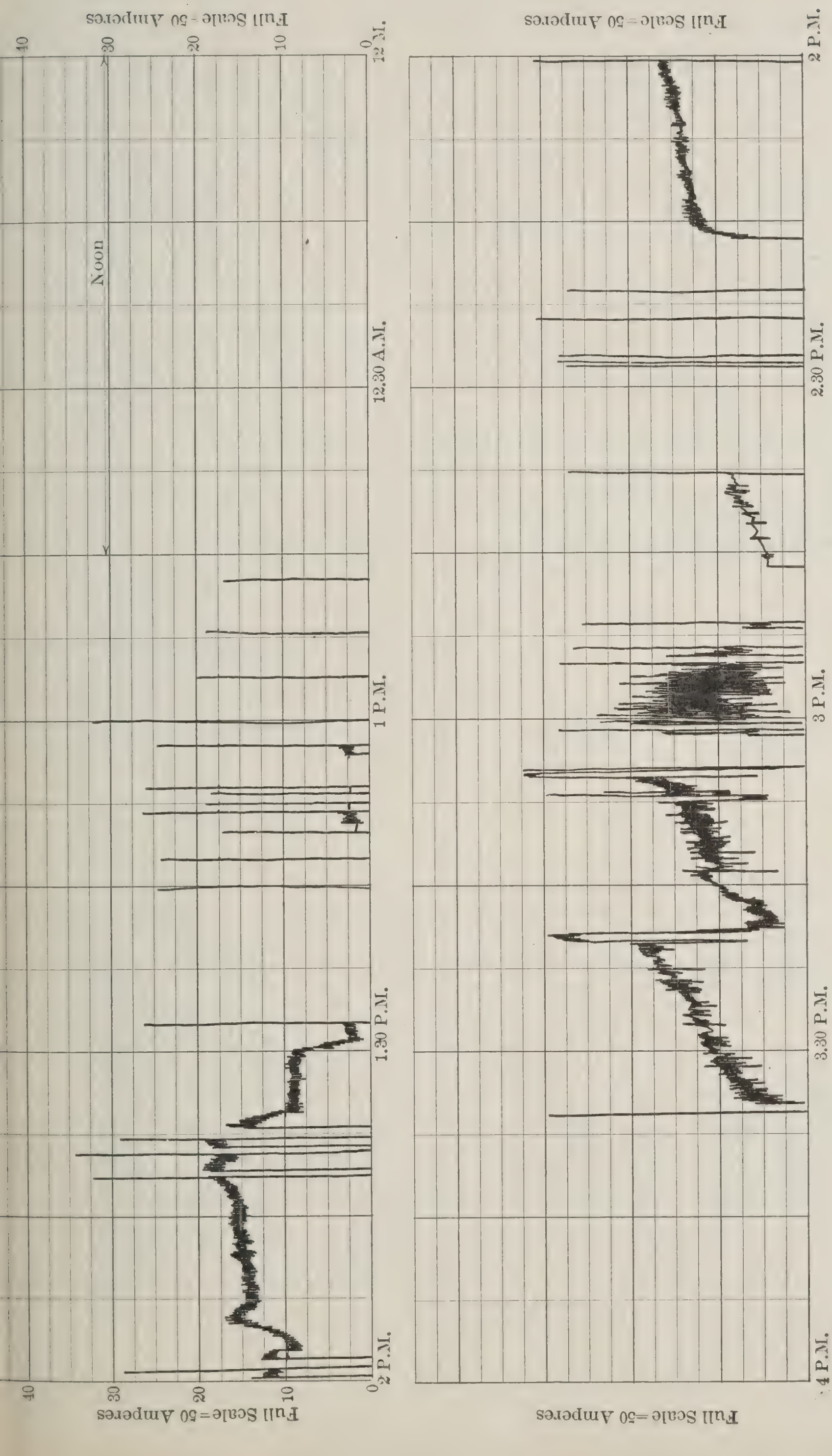


Fig. 2 Complete Meter Record for One Day from 72-in. Boring Mill

TABLE 2 ANALYSIS OF A COUNTERBORING OPERATION

	Number of Hole						Total Time To Drill	Adjustment	Change	Complete
	1	2	3	4	5	6				
Min.	7.3	7.5	7.4	7.5	7.4	7.5	44.6	19.6	8.2	72.4
% of Total							61.5	27.2	11.3	100
Min.	7.1	7.4	7.3	7.4	7.4	7.4	44	17.7	8.4	70.1
% of Total							62.8	25.2	12.0	100
Min	7.5	7.5	7.5	7.3	7.4	7.5	44.7	15.5	9.9	70.1
% of Total							64	22	14	100
Min	7.3	7.1	6.8	7.0	7.1	7.5	42.8	24.2	7.5	74.5
% of Total							57.5	32.5	10	100

No. load of drill, 1.75 kw. Power to drill, 1.75 kw. Average running load, 2.8 kw.

TABLE 3 ANALYSIS OF DRILLING AND TAPPING OPERATION

Set up equals 56 Min.

Pole No.	Operation	Time Required, Minutes Hole No.						Total Time Required			
		1	2	3	4	5	6	To Drill	To Adjust Drill	To Change Drill	Complete
1	$1\frac{9}{16}$-in. drill	1.1	1.3	1.1	1.3	1.3	1.3	7.4	5.6	....	13
	$1\frac{9}{32}$-in. drill	4.5	4.2	4.4	4.3	4.2	6.6	28.2	6.0	8.3	42.5
	$1\frac{5}{16}$-in. tap	0.9	0.6	0.7	0.7	0.5	0.8	4.2	13.2	6.6	24
							Total min.	39.8	24.8	14.9	79.5
							% of total	50	31.2	18.8	100
Turn Over = 26.2 min. delay for drawing = 31.4 min.; other delay = 57											
2	$1\frac{9}{16}$-in. drill	1.4	1.3	1.3	1.1	1.2	1.1	7.4	6.3	....	.....
	$1\frac{9}{32}$-in. drill	5.0	4.8	4.8	4.6	4.9	4.6	28.7	8.2	....	.....
	$1\frac{5}{16}$-in. tap	0.9	1.1	1.1	1.1	1.1	1.1	6.4	30.5	12.5	100
							Total, min.	42.5	45.0	12.5	100
							% of total	42.5	45.0	12.5	100
Turn Over = 21.2											
3	$1\frac{9}{16}$-in. drill	1.7.	1.3	1.5	1.2	1.5	1.2	8.4	4.4	....	.....
	$1\frac{9}{32}$-in. drill	4.9	5.1	5.6	5.4	5.4	4.7	31.1	......	3.8	.....
	$1\frac{5}{16}$-in. tap	0.6	0.8	0.6	0.8	0.8	0.6	4.2	14.3	8.4	.....
							Total min.	34.7	18.7	12.2	65.6
							% of total	53	28	19	100

DRILLING AND TAPPING

6 Table 3 is from a record made while an 8-pole revolving field was drilled and tapped for fitting pole shoes. To place the job in position required 56 min., owing to lack of prompt crane service. No lining up was required; the work was simply set upon the bed plate of the radial drill.

7 The total time consumed in adjusting the drill to proper position per pole was 5.6 min. The $1\frac{9}{16}$-in. holes were first drilled $\frac{3}{8}$-in. deep, requiring about 1.2 min. each, a total time for drilling the six holes of 7.4 min. The $1\frac{9}{16}$-in. drill was then removed and replaced by a $1\frac{9}{32}$-in. drill, requiring about 8 min. To drill each $1\frac{9}{32}$-in. hole $2\frac{1}{2}$ in. deep took about $4\frac{1}{2}$ min., or 28 min. for the six holes. Adjustments of the drill took 6 min., making a total time of $42\frac{1}{2}$ min. to complete drilling the six $1\frac{9}{32}$-in. holes. The drill was then removed and a $\frac{5}{16}$-in. tap substituted, this change requiring 6.5 min. To tap each hole took from 0.6 to 0.9 min., making a total of 4.2 min. for the six holes. The time taken for adjustments was 13.2 min., making a total of 24 min. for the tapping operation. The actual cutting time for tapping was $17\frac{1}{2}$% of the total time. The total cutting time per pole, including one tapping and two drilling operations, was 50% of the total completing time, 31.2% of the time being consumed in adjusting, and 18.8 % in changing drills for taps. About 20 to 30 min. were consumed in waiting for a crane to turn over the job in order to drill the next pole piece. The time required to complete the job can be analyzed as follows:

Set up and wait for crane	56 min.
Complete poles	560 min.
Turn over (crane service)	240 min.
Total	856 min., or 14 hr. 16 min.
Total cutting time	320 min., or 5 hr. 20 min.
Time factor	38%

RECORDS FROM 5-FT. RADIAL DRILL

8 The following results were obtained by taking records on a 5-ft. radial drill driven by a $7\frac{1}{2}$-h.p. adjustable-speed motor, with rated speeds ranging from 400 to 1600 r.p.m., while drilling a series of holes in a large steel casting. Out of $11\frac{1}{2}$ hr., 4 hr. 42 min. were consumed in actual drilling, the other 6 hr. 48 min. being required to make adjustments. In this case, the time factor was 41%. The average running load while drilling was 1.5 kw., making an average daily load of 0.7 kw., or a load factor of 10%.

9 While drilling a series of 22 holes, 67.5% of the time was consumed in actual drilling, the remaining 32.5% being consumed in moving the drill from one position to the next. In another case where holes were to be drilled and tapped, 74.5% of the time was consumed in actual drilling; while in tapping the holes, the machine was in use 44% of the time, the remainder being consumed in making adjustments. Records taken while a series of small jobs were drilled show that the time factor was as low as 20%, much time being lost in obtaining drawings and auxiliary materials.

PORTABLE MILLING MACHINE

10 On a portable milling machine driven by an induction motor of 3 h.p., 720 r.p.m., while milling slots and dovetails in an iron casting, the average running load was 1.5 kw. to 2 kw., giving a load factor of from 50 to 67%. The time factor was 54%. 24% of the total time was required to make adjustments of the cutting tool; the remaining 22% was required to set up the job, that is, to place the portable machine in a central position upon a table inside the circular casting where the machine could be completely revolved and clear the inside of the frame.

PORTABLE SLOTTER

11 On a portable slotter driven by a 10-h.p., 720-r.p.m. motor while cutting slots in a cast-iron frame, the arm carrying the cutting tool is moved up and down by reduction gearing and a rack. The cut is taken on the up-stroke. The record (not printed) shows that the peak load occurs just before the cut is taken, i.e., when the arm is reversed for the upward motion; the minimum load occurs on the downward stroke. The record also shows a variation in the amount of power required to produce the cut. This variation is due to irregularities in the feeding mechanism on the machine tested, a ratchet which had become worn. The time factor was 50% and the load factor 12%.

COMPARISON OF MILLING AND SLOTTING

12 From the results obtained an interesting comparison can be drawn between the time required to cut slots with a miller and with a slotter, as shown in the following table:

	Size of Slots Inches	Cutting Time Minutes	Minutes to Cut 1 In.
Miller	$7\frac{1}{8} \times \frac{5}{8} \times 12\frac{1}{2}$	11.8	0.95
Slotter	$7\frac{1}{8} \times \frac{5}{8} \times 15\frac{3}{4}$	8.4	0.53

13 The results show that the actual cutting time per inch of the slotter is but 56 %of the time of the miller; both were removimg exactly the same amount of material. The curves also show that the intervals required for adjustment of the positions of the tools from one slot to another averaged 6.1 min. on the miller and 3.1 min. on the slotter, an advantage of 50% again in favor of the slotter. The time to set up the work must be included in order to determine the relative advantage of one machine over the other. The setting-up time was found to depend more upon the work than upon the tool, and neither tool had an advantage. Two hours were required to set up the job on each tool. The results may then be summarized by comparing the operations of the machines in cutting two similar jobs of 12 slots 10 in. long:

	Setting up Time	Adjustments	Cutting Slots	Total
Miller	2 hr.	1 hr. 12 min.	1 hr. 44 min.	4 hr. 56 min.
Slotter	2 hr.	0 hr. 36 min.	1 hr. 04 min.	3 hr. 40 min.

14 This shows that the total time required by the slotter was but 74% of that required by the miller; a saving of 1 hr. 16 min. on every such job.

POWER REQUIRED TO OPERATE PLANERS

15 Table 4 contains a summary of results to determine the power required to operate various motor-driven planers. The average time factor in ordinary planing operations is about 50 to 60%.

CURVES FROM MACHINES IN A STEEL TUBE MILL

16 Fig. 3 shows a curve taken from a motor operating welding rolls while lap welding 5-in. tubes. The rolls were driven by a 150-h.p. induction motor.

TABLE 4 ANALYSIS OF POWER REQUIRED TO OPERATE PLANERS

Size Planer	H. P. of Motor	Reversal				Cutting Stroke		Reverse Stroke		No Load	Average Running Load		Load Factor at 50% Time Factor
		a		b									
		Kw.	% Full Load	Kw.	% Full Load	Kw.	% Full Load	Kw.	% Full Load		Kw.	% Full Load	
56 in. x 12 ft.	15	8.5	56	6.5	43	2.5–5	17–34	3	20	2	4–5	30	15
7 ft. x 12 ft.	15	6	40	3	20	1–2	7–14	0.75	5	0.6	4	25	12.5
14 ft. x 20 ft.	30	34	113	30	110	6–10	20–34	8	27	5.5	11	35	17
10 ft. x 20 ft.	50	22	45	16	30	12–16	25–32	7	14	4.5	16	30	15
14 ft. x 30 ft.	40	40	115	24	60	10–16	25–40	10	25	5	15	36	18

Reversal *a*, from cutting to return stroke.
Reversal *b*, from return to cutting stroke.

To reduce the peak load thrown on the motor, the rolls were equipped with a 5-ft. diameter, 5000-lb. fly wheel. This record is interesting, in that it shows that a friction load of 12 kw. was required about 91% of the time, under which condition of operation the motor, on account of its light load, was operating at a power factor of about 30%, which is an undesirable condition for the power plant.

17 The duration of peak load is in each instance about eight seconds, which amounts to only about 9% of the total cycle of operation. During this period the input to the motor was from 128 to 160 kw. A study of the records shows that a smaller motor should be installed. The motor should be designed with a larger slip, or drop in speed between no load and full load, so that with a some-

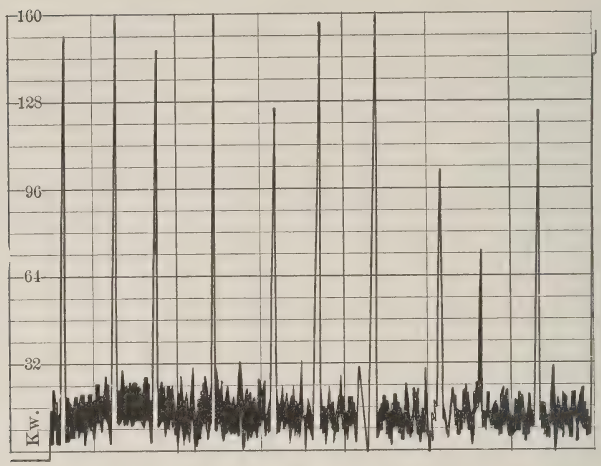


FIG. 3 RECORD FROM WELDING ROLLS EQUIPPED WITH FLYWHEEL

what larger flywheel, when the load is steadily thrown on the motor, it would slow down, allowing the fly wheel to give forth energy and in this way moving out or lowering the peaks for instantaneous demand for current from the line. With a smaller motor the power factor would be increased, the efficiency improved, and the load factor also improved. At the time the tests were made a meter was used, having a paper speed of 24 in. per hr. The record shows that tubes were rolled at the rate of 40 per hour.

FIG. 4 RECORD FROM HOT-ROLL SCARFER

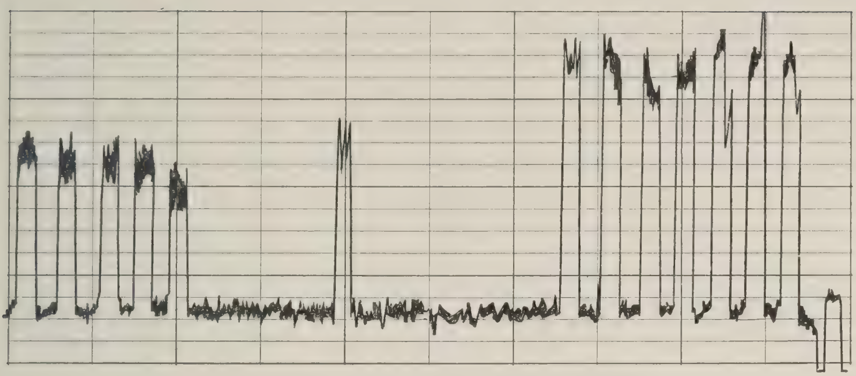


FIG. 5 RECORD FROM SHEAR SCARFER

18 The curve represented by Fig. 4 was taken from a 75-h.p. motor, operating a hot-roll scarfer, scarfing sheets for 12-in. lap-welded tubes. It shows that the friction load of 18 kw. is practically constant for about 90 % of the time, and that the maximum or peak load of about 34 kw. occurred for about 4% of the total time. At the time these records were taken the meter was operating at a speed of 24 in. per hr., and the rolls turning out 10 and 15 tubes per hour. The difference in peak load is due to the fact that an increase in width of the metal causes a slight increase in power. Undoubtedly a 50-h.p. motor would have been satisfactory for this work with much more economy than the installed motor.

19 The curve represented by Fig. 5 is a record showing the operating conditions of a 50-h.p. induction motor driving a shear scarfer. On this curve the

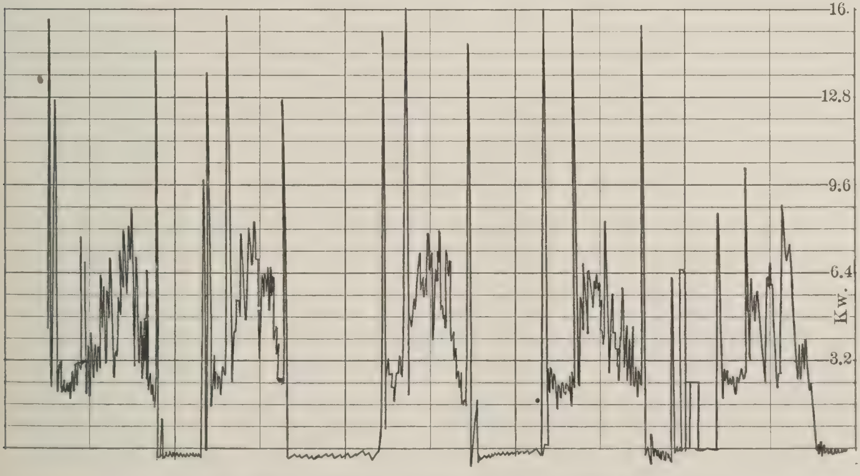


FIG. 6 RECORD FROM PIPE CUTTING-OFF MACHINE

friction load was shown at about 2.3 kw., of a duration of 55% of the time. The peak loads require from 10 to 13 kw. for the balance, or 45% of the time. As the duration of the peak is only about 34 seconds, a smaller motor of about 25 h.p. would be of sufficient capacity to do the work. In this instance, as in the others, a meter having a 24-in. movement of the record was used.

PIPE CUTTING-OFF MACHINE

20 The record shown in Fig. 5 was taken from a 5-h.p. motor operating a pipe cutting-off machine, cutting 18-in. tubes at an average cutting speed of about 38 ft. per min. It will be noted that about 16 kw. were required for starting, while the average load during cutting was about 6 kw. The motor was reversed for the reaming operation, and the peak was very large, going off the scale. To start the machine the motor was again reversed, such manipulation causing very severe overloads on the motor and the gearing to the machine. For a reversing operation of this nature an induction motor having a large slip would be desirable, or a slip-ring type of motor, thus reducing the demand upon the line. A direct-current motor, if used, should be supplied with very heavy compound winding.

APPENDIX NO. 2. POWER REQUIRED TO REMOVE METAL

1 The power required to remove metal depends upon the nature of the cutting tool and the amount of metal removed per minute. Cutting tools may be divided into three general classes: (*a*) lathe tool type; (*b*) drills; (*c*) milling cutters.

LATHE TOOL TYPE

2 The lathe tool is used on lathes, boring mills, planers, shapers and slotters. Tests show that the power required by a tool of this kind when removing metal depends upon the cutting angle of the tool and the number of cubic inches of metal removed per minute. From observation and data

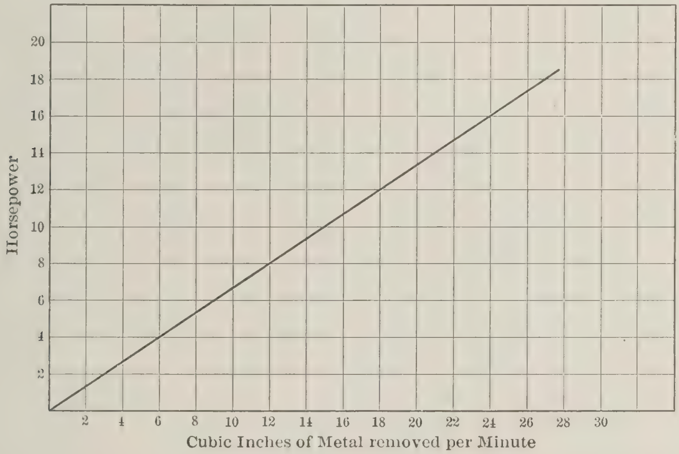


FIG. 1 RELATION BETWEEN HORSEPOWER AND CUBIC INCHES METAL REMOVED; MILD STEEL, 0.40% CARBON

obtained by means of the graphic recording meter, and the use of tools having a cutting angle of about 75 deg. to 80 deg. the curve shown in Fig. 1 was obtained. The results were independent of the cutting speed, feed and depth of cut, and show that a definite relation exists between the horsepower required to remove metal and the number of cubic inches removed per minute. The cubic inches of metal removed per minute were found to be as follows:

(*a*) area of cut (sq. in.) $\times$ cutting speed (ft. per min.) $\times$ 12

(*b*) area of cut (sq. in.) = depth of cut (in.) $\times$ feed (in. per revolution)

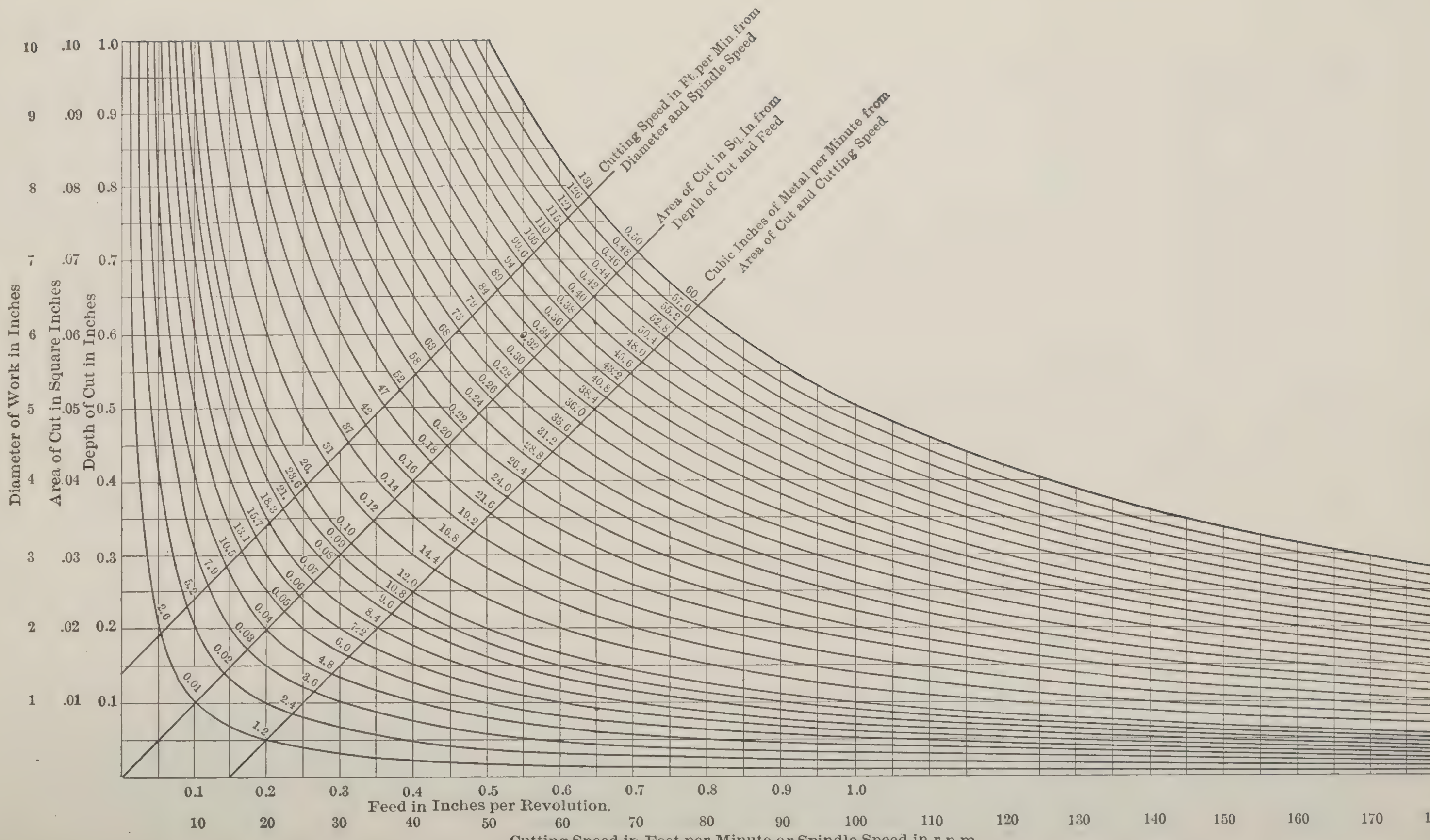


PLATE 1 MACHINE TOOL CALCULATOR FOR LATHES, PLANERS, SHAPERS, SLOTTERS AND BORING MILLS

DIRECTIONS FOR USING PLATE 1

a To find cutting speed: From intersection of horizontal line corresponding to diameter and vertical line corresponding to spindle speed, follow nearest curve and use value found in oblique line of figures marked cutting speed.

b To find area of cut: From intersection of horizontal line corresponding to depth of cut and vertical line corresponding to feed, follow nearest curve and use value found in oblique line of figures marked area of cut.

c To find cubic inches of metal removed per minute: From intersection of horizontal line corresponding to area of cut and vertical line corresponding to cutting speed follow nearest curve and use value found in oblique line of figures marked cubic inches of metal removed per minute.

To use curve, knowing diameter of work, spindle speed, depth of cut and feed, find cutting speed from (*a*) area of cut from (*b*) and cubic inches of metal removed per minute from (*c*).

3 The horsepower required to remove metal with the tools ordinarily employed can be expressed by:

$$\text{h. p.} = \text{a constant} \times \text{cu. in. removed per min.}$$

The constant varies with the kind of metal removed.

4 In order to estimate the amount of power required to remove a given amount of metal per minute the graphic method shown in Plate 1 has been designed. This diagram is a multiplication table; those familiar with analytical geometry will recognize the equilateral hyperbola whose equation, referred to its asymptotes, is xy = constant.

5 To determine the cutting speed the usual procedure is as follows:

$$\text{cutting speed (ft. per min.)} = \frac{\pi \times \text{diameter} \times \text{r.p.m.}}{12}$$
$$= \text{constant} \times \text{diameter} \times \text{r.p.m.}$$

In the diagram each hyperbola corresponds to a given cutting speed. The coördinates of all diameters and spindle speeds producing the same speed intersect on the same hyperbola. The cutting speed corresponding to any diameter, rotation at any number of r.p.m., is found indicated on the hyperbola passing through the intersection of the coördinates corresponding to the given values of diameter and r.p.m.

6 In a similiar manner an area corresponding to any depth of cut in inches and feed in inches is obtained, and also the cubic inches of metal removed per minute can be determined from the area of cut and the cutting speed. The directions for using the diagram are given in connection with it.

7 With the cutting tools ordinarily employed the following values have been found by tests to exist for the horsepower required to remove 1 cu. in. of the following metals, per min.:

Brass and similar alloys	0.2 to 0.3
Cast iron	0.3. to 0.5
Wrought iron	0.6
Mild steel (0.30%–0.40% carbon)	0.6
Hard steel (0.50% carbon)	1.00 to 1.25
Very hard tire steel	1.50

8 It must be remembered that these constants represent general average conditions; considerable variation may occur where special cutting tools are used and special grades of metal are encountered.

LATHES

9 The following examples will explain the application of the diagram, Plate 1, to lathe work.

Example:	Diameter of work	= 5.5 in.
	Spindle speed	= 45 r.p.m.
	Depth of cut	= 0.45 in.
	Feed per revolution	= 0.06 in.

10 Find the intersection of the horizontal line through 5.5 in. diameter of work, and the vertical line through 45 r. p. m. spindle speed. The curves pas-

sing nearest this intersection correspond to a cutting speed of 63 and 68 ft. per min., indicating by interpolation a cutting speed in this case of 65 ft. per min. The area of cut, with depth of cut 0.45 in. and feed 0.06 in. is 0.027 sq in. The cubic inches of metal removed per minute, corresponding to an area of cut 0.027 sq. in. and a cutting speed of 65 ft. per min., is determined by finding the intersection of the horizontal line passing through 0.027 sq. in. area of cut and 65 ft. per min. This intersection is between the curves corresponding to 19.2 and 21.6 cu. in., showing that about 20 cu. in. of metal are removed per min. If the metal removed is wrought iron, the horsepower required is $0.6 \times 20 = 12$ h.p. If 0.50% carbon steel is turned, $1.00 \times 20 = 20$ h.p., is required. Brass would require $0.25 \times 20 = 5$ h.p.

BORING MILL

Example:	Diameter of work	=45 in.
	Speed of table	= 4.5 r.p.m.
	Depth of cut	= 0.25 in.
	Feed	=0.10 in. per revolution

11 The diameter of work goes only to 10 in. in the vertical column of the diagram. These may be multiplied by 10, and if used with the spindle speeds as they stand, the results in the oblique column of cutting speeds must be multiplied by 10. In case of large diameters the spindle or table speeds are usually low. The simplest way to use the diagram in these cases is to interchange *diameter of work and spindle speed,* i. e., assume that the diameter of the work is 10, 20, 30, etc., in the horizontal column, and the table speed under 1, 2, 3, etc., in the vertical column. In the problem under consideration the cutting speed is as follows:

12 The intersection of the horizontal line through 4.5 and the vertical line through 45 correspond to a cutting speed of 52 ft. per min. The area of cut is 0.025 sq. in. The intersection of the horizontal line through 0.025 sq. in. area of cut, and the vertical line through 52 ft. per min. cutting speed lies between curves representing 14.4 and 16.8 cu. in., indicating that 15 cu. in. are removed per min. If cast iron of a soft quality is removed the power required for cutting will be $15 \times 0.3 = 4.5$ h. p. If the cast iron is of hard quality, $0.5 \times 15 = 7.5$ h. p., will be required.

SHAPER OR PLANER

Example:	Depth of cut	= 0.75 in.
	Feed per stroke	= $\frac{1}{16}$ in.
	Cutting speed	= 45 ft. per min. (from characteristic of planer or shaper)

Area of cut $0.75 \times \frac{1}{16} = 0.046$ sq. in.

13 The cubic inches of metal removed per minute, corresponding to an area of cut of 0.046 sq. in., and a cutting speed of 45 ft. per min., is 24. The power required for cutting in the machine a hard grade of cast iron will under these conditions be $24 \times 0.5 = 12$ h. p.

14 In a planer the power required for reversing is usually considerably more than that required to cut metal, depending upon the design of the reversing

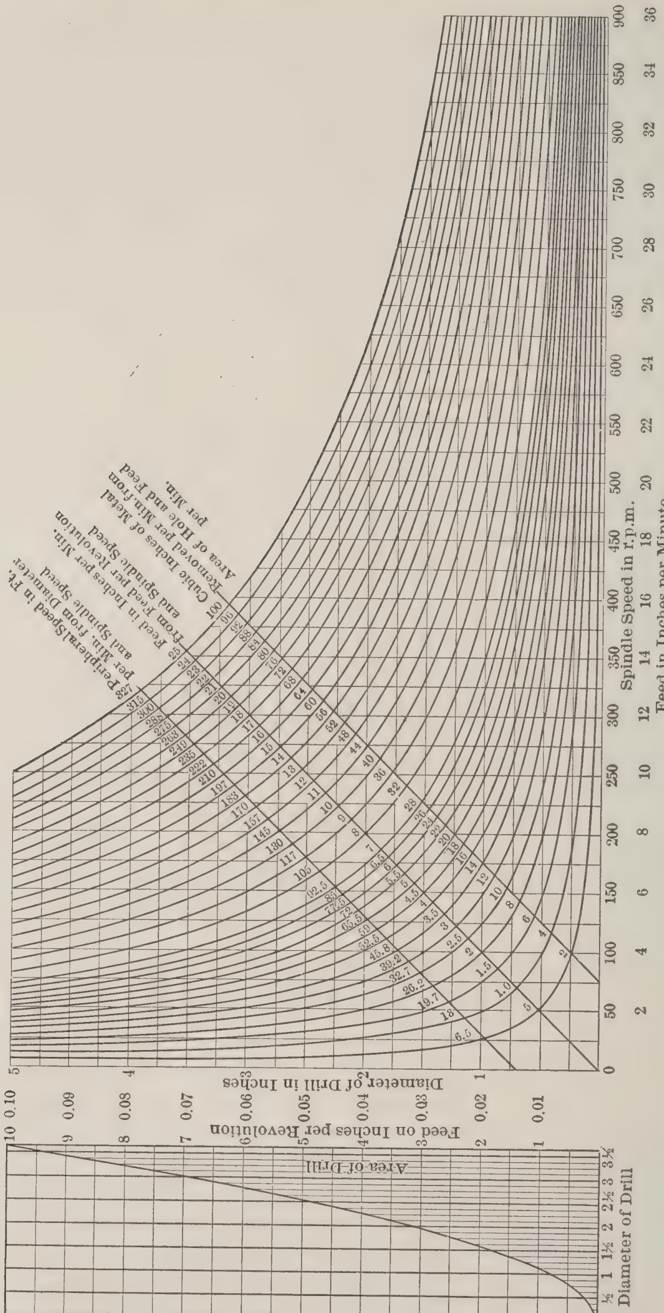


PLATE 2 MACHINE TOOL CALCULATOR FOR DRILLS

DIRECTIONS FOR USING PLATE 2

a To find cutting speed: From intersection of horizontal line corresponding to diameter of drill and vertical line corresponding to spindle speed, follow nearest curve and use value found in oblique line of figures marked cutting speed.

b To find feed in inches per minute from feed per revolution and spindle speed: From intersection of horizontal line corresponding to feed in inches per revolution and vertical line corresponding to spindle speed follow nearest curve and use value found in oblique line of figures marked feed in inches per minute.

c To find area of drill from diameter of drill use curve on left side of figure: Find intersection of vertical line corresponding to diameter of drill with the curve follow the horizontal line passing through this intersection and obtain area under area of drill in vertical column.

d To find cubic inches of metal removed per minute: From intersection of horizontal line corresponding to area of drill and vertical line corresponding to feed per minute follow nearest curve and use value found in oblique line of figures marked cubic inches of metal removed per minute.

Knowing diameter of drill, spindle speed and feed per revolution, find cutting speed from (*a*), and cu. in. metal removed per minute from (*b*), (*c*), and (*d*).

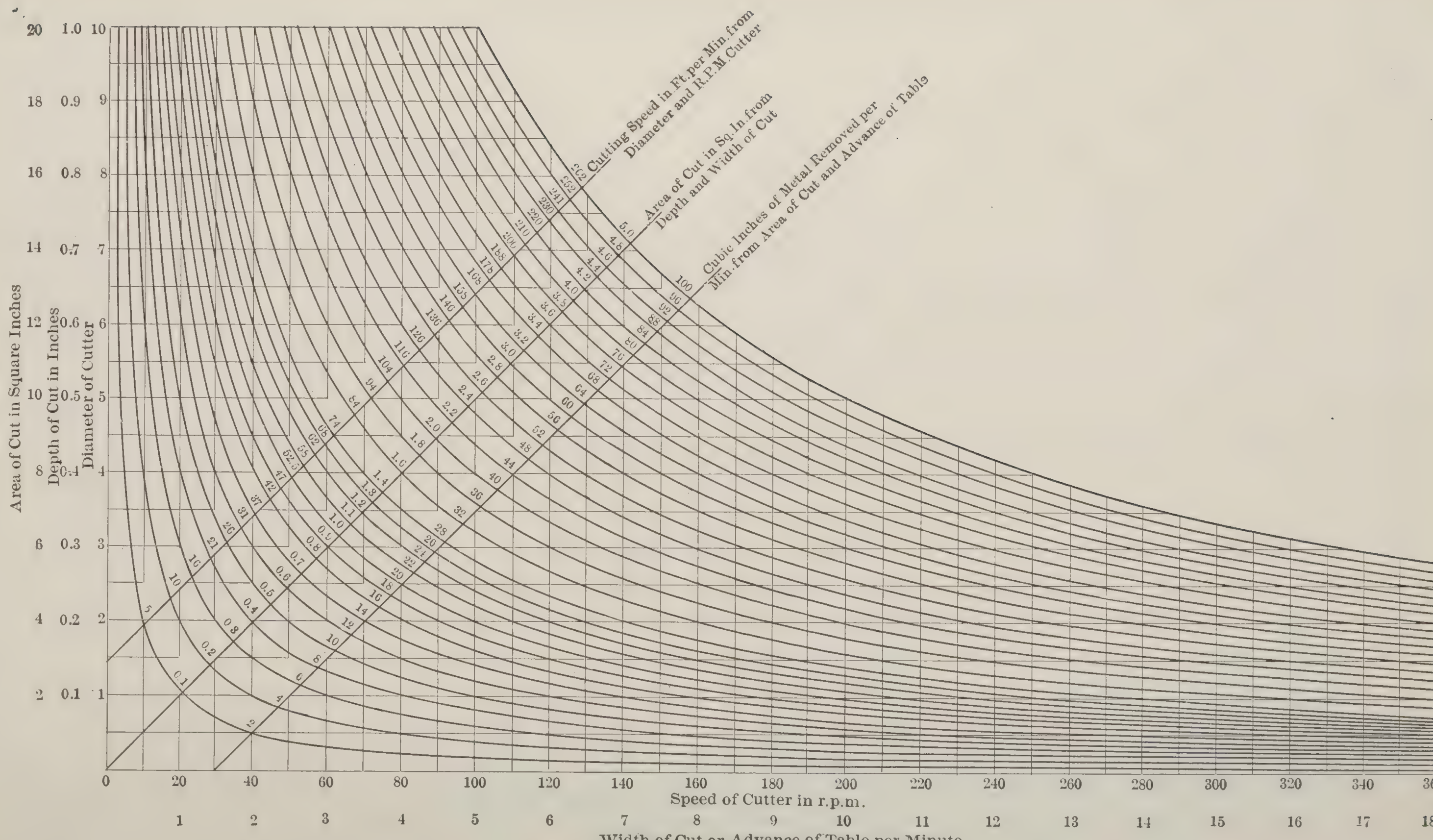


PLATE 3 MACHINE TOOL CALCULATOR FOR MILLING MACHINES

DIRECTIONS FOR USING PLATE 3

a To find cutting speed: From intersection of horizontal line corresponding to diameter and vertical line corresponding to spindle speed of cutter, follow nearest curve and use value found in oblique line of figures marked cutting speed.

b To find area of cut: From intersection of horizontal line corresponding to depth of cut and vertical line corresponding to width of cut, follow nearest curve and use value found in oblique line of figures marked area of cut.

c To find cubic inches of metal removed per minute: From intersection of horizontal line corresponding to area of cut and vertical line corresponding to advance of table per minute, follow nearest curve and use value found in oblique line of figures marked cubic inches of metal removed per minute.

To use curve, knowing the diameter of cutter, spindle speed, depth of cut, width of cut, and advance of table per minute, find cutting speed from (*a*), area of cut from (*b*), cubic inches metal removed per minute from (*c*).

mechanism, the flywheel effect and the speed characteristic of the motor. In a shaper the power required to reverse is not very great, and is usually less than the power required for cutting.

SLOTTER

15 In most cases the cutting tool is fed inwardly on this type of machine; the following example shows how the diagram is used to determine the rate of removing metal. With other methods of feeding the tool the diagram is used in the same way as in the case of a planer or a shaper.

Example:	Width of tool and cut	= 0.5
	Feed per stroke	= 0.06
	Cutting speed	= 35 ft. per min.
	Area of cut 0.5 x 0.06	= 0.03 sq. in.

16 The cubic inches of metal removed per minute from the intersection of the horizontal and vertical line through 0.03 sq. in. and 35 ft. per min. are 13. In the case of mild steel the horsepower required would be 13 x 0.6 = 7.8 h. p.

DRILLS

17 The power required in drilling operations can also be expressed as a constant times the cubic inches of metal removed per minute. The conditions are, however, more complicated than in the lathe tool, since the friction of the drill and the chips on the sides of the hole increase the power requirement as the drill enters the metal. This is especially true when cast iron is drilled, as chips have a jamming action. The variable cutting speed at the cutting edge of the drill, from zero at the center to the peripheral speed of the drill, also causes a jamming action and tends to increase the power per cubic inch per minute over that required to remove the same amount of metal by means of the lathe tool type. With drills generally employed, the value per horse power per cubic inch of metal removed per minute. is about double that required by ordinary lathe tools.

18 Plate 2 is a diagram with full instructions for determining the cubic inches of metal removed with drills. The constants for determining the power required are about double those for lathe tools.

Example:	Size of drill	= 2 in. diameter
	Feed per minute	= 2.5 in.
	Speed of drill	=150 r.p.m.
	Metal drilled: cast iron.	

19 The peripheral or maximum cutting speed of the drill is found as follows (Rule *a*, Plate 2): The horizontal line corresponding to a diameter of 2 in. intersects the vertical line corresponding to 150 r.p.m. on the curve corresponding to a cutting speed of 77.5 ft. per min. The area of the 2 in. drill (rule *c*) is 3 sq. in. This area at a feed of 2.5 in. per min. corresponds to removing 7 cu. in. per min. (rule *d*). For cast iron the horsepower per cu. in. per min. is about 0.8, twice that for lathe tools, hence the power required to drive the drill in this case is 0.8×7 = 5.6 h.p., which agrees closely with an actual test. For

mild steel the power required is $1.2 \times 7 = 8.4$ h.p. In drilling a hole of this size the friction of the chips does not increase the power materially as the depth of the hole increases, since there is sufficient space for the drill to free itself of chips.

MILLING CUTTERS

20 Plate 3 is a diagram with full instructions for determining the amount of metal removed per minute by a milling machine.

Example:	Width of cut	= 8 in.
	Depth of cut	= 0.2 in.
	Advance of table per min.	= 5 in.
	Area of cut is 8×0.2	= 0.16 sq. in.

21 To find the cubic inches of metal removed per minute, find on the diagram the intersection of the horizontal line through 0.16 sq. in., and a vertical line corresponding to a table advance of 5 in. per min. The curve passing through this intersection corresponds to a rate of cutting of 16 cu. in. of metal per min. For machinery steel or mild steel, the power required by a horizontal milling machine of this type is about 1.6 per cu. in. per min, making the total requirement $1.6 \times 16 = 25.6$ h.p. A vertical miller requires about 1 h.p. per cu. in. per min., or 16 h.p. under the foregoing conditions.

22 The power required by milling cutters varies according to their construction, and care should be employed to determine the proper constant for each class of cutters. By means of tests made with the graphic meter on motor-driven tools the proper constant can easily be determined in any given case.

APPENDIX NO. 3. SIZES OF MOTORS RECOMMENDED TO DRIVE MACHINE TOOLS

The accompanying tables contain the sizes and speeds of motors usually employed with the average duty indicated for machine tools. The constant speed motors are selected with a view to utilizing speeds as near as possible to those obtainable with 60-cycle induction motors. By this means the same gear ratios can be employed with either direct current motors or 60 cycle induction motors.

2 The average load factor for motors driving lathes is from 10 to 25 %. On some special machines, as driving wheel and car wheel lathes, the cuts are all heavy, which increases the average load factor to from 30 to 40%.

3 For extension boring mills, 5 h.p. motors are used to move the housings on from 10 ft. to 16 ft. mills, 7½ h.p. for from 14 ft. to 20 ft. mills and 10 h.p. for from 16 ft. to 24 ft. mills. The load factor of the driving motor on boring mills averages fom 10 to 25 %.

4 The load factor of motor-driven drills is about 40%, when the larger drills applicable thereto are used. If the smaller drills are used the load factor averages 25% and lower.

5 For the average milling operations the load factor averages from 10 to 25 %. On slab milling machines where large quantities of metal are renewed it will average from 30 to 40%.

6 The work on this class of machinery is usually light and much time is required in making adjustments. Hence the load factor is rarely higher than 20%.

7 On planers the load factor averages between 15 and 20%. The motor must be large enough to reverse the bed quickly, yet this peak load occurs for such short intervals that it does not increase the average load per cycle very much.

8 The work done on shapers is of a varying character. With light work the load factor will not exceed from 15 to 20%; with heavy work, the load factor will be as high as 40%.

9 The conditions encountered on slotters are similar to those on shapers.

TABLE 1 SIZES AND SPEEDS OF MOTORS ON LATHES

ENGINE LATHES

Adjustable speed, ratio 1 : 3.

Swing In.	Light Duty			Medium Duty			Heavy Duty		
	h.p.	Adjst. Speed	Const. Speed r.p.m.	h.p.	Adjst. Speed	Const. Speed r.p.m.	h.p.	Adjst. Speed	Const. Speed r.p.m.
14	2	Ratio	1800	3	Ratio	1800	5	Ratio	1200
16	3	1 : 3	1800	5	1 : 3	1200	5	1 : 3	1200
18–20	3		1800	5		1200	7 : 5		1200
22–24	5		1200	7 : 5		1200	10		1200
27–30	7½		1200	10		1200	15		1200
36–48	7½		1200	10		1200	20		900

SPECIAL LATHES

Type	h.p.	Adjustable Speed
Car wheel 48 in.	20	1 : 3
Double axle, moderate duty	15	1 : 3
Heavy duty	25	1 : 3

DRIVING WHEEL LATHES

Size, in.	h.p.	Adjustable speed
51	15	Ratio 1 : 3
60–69	20	
79	25	1200 r.p.m.
84	25	
90	30	
100	50 5 tail stock	

TABLE 2 SIZES AND SPEEDS OF MOTORS
Vertical Boring Mills

Size, in.	h.p.	Adjustable Speed	Constant Speed
24–30 in.	5	Ratio 1 : 3	1200
36–42 in.	7½		1200
60–90 in.	10 5–rail		1200
100 in.	15 5–rail		1200
10 ft.	20 7½–rail		900
12 ft.	20 7½–rail		900
14 ft.	25 7½–rail		900
16 ft.	30 10–rail		900

TABLE 3 SIZES AND SPEEDS OF MOTORS ON DRILLS
Radial Drills

Size, ft.	h.p.	Adjustable speed	Constant Speed
4	3	Ratio 1 : 3	1800
5	5		1200
6	5		1200
10	7½		1200

Upright Drills

Size, in.	h.p.	Adjustable Speed	Constant Speed r.p.m.
Friction	½	Ratio 1 : 3	1800
15	½		1800
20–26	1		1800 1200
28–34	2		1800 1200
42–50	3		1800 1200

Multiple-Spindle Drills

Size, in.	h.p.	Adjustable Speed	Constant Speed
4–2	7½	Ratio 1 : 3	1200
6–2	10		1200
8–2	10		1200

TABLE 4 SIZES AND SPEEDS OF MOTORS ON MILLING MACHINES

Horizontal—Plain or Universal

Table Feed In.	Cross Feed In.	Vertical Feed In.	h.p. Mod. Heavy	Adjustable Speed	Constant Speed r.p.m.
24	8	18	3	Ratio 1 : 3	1800
30	10	18	5– 7½		1200
36	12	20	7½–10		1200
50	12	20	10–15		....

Vertical Milling Machines

Table Diameter in.	Spindle Diameter in.	h.p.	Adjustable Speed	Constant Speed r.p.m.
28	4	5	Ratio 1 : 3	1200
32	4	7½		1200
40	4½	10		1200
54	5	15		1200
70	6	20		900

Slab Milling Machines

Width of Table, in.	h.p.	Adjustable Speed	Constant Speed r.p.m.
24–30	10	Ratio 1 : 3	1200
36	15		1200
60	25		900
36 heavy	25		900
42 heavy	50		900

TABLE 5 SIZES AND SPEEDS OF MOTORS

Horizontal Boring, Drilling and Milling Machines

Spindle, in.	h.p.	Adjustable Speed	Constant Speed r.p.m.
3½	3	Ratio 1 : 3	1800
4	5		1200
5	7½		1200
6	10		1200
7	15		1200

TABLE 6 SIZES AND SPEEDS OF MOTORS ON PLANERS

Medium Duty			Heavy Duty		
Size, in.	h.p.	Constant Speed r.p.m.	Size, in.	h.p.	Constant Speed r.p.m.
24 x 24	5	900	24 x 24	7½	900
30 x 30	7½	900	42 x 42	25	900
36 x 36	10	900	56 x 56	25	900
48 x 48	15	900	Frog and	30	900
56 x 56	15	900	Switch Forge	...	...
			12 x 10 ft.	60	720
			14 x 12 ft.	10 (rail)	...
				75	...
				12 (rail)	720

TABLE 7 SIZES AND SPEED OF MOTORS ON SHAPERS

Size In.	h. p.	Adjustable Speed	Constant Speed r. p. m.
14–20	3	Ratio 1:3	1800
24	5		1200
36	7½		1200

TABLE 8 SIZE AND SPEEDS OF MOTORS ON CRANK SLOTTERS

LIGHT, MEDIUM AND HEAVY

Size In.	h. p.	Medium	Adjustable Speed	Constant Speed r. p. m.
10	3	5	Ratio 1:3	1800
10–16	5	7½		1200
20	7½	10		1200
26–30	15	..		1200
GEARED SLOTTERS				
24–60	20	....	1:3	900

TABLE 9 SIZES AND SPEEDS OF MOTORS ON COLD SAWS

Diameter In.	Thickness In.	h. p.	Adjustable Speed	Constant Speed r. p. m.
12	$\frac{5}{32}$	2	Ratio 1:3	1800
15	$\frac{5}{32}$	2		1800
18	$\frac{3}{16}$	3		1800
20	$\frac{3}{16}$	3		1800
24	$\frac{3}{16}$	5		1200
32	$\frac{3}{16}$	$7\frac{1}{2}$		1200
36	$\frac{3}{16}$	10		1200

TABLE 10 SIZES AND SPEEDS OF MOTORS ON GRINDERS

Size In.	h. p.		Constant Speed r. p. m.
	Medium	Heavy	
10x 50	5	$7\frac{1}{2}$	1200
10x 72	5	$7\frac{1}{2}$	1200
10x 96	5	$7\frac{1}{2}$	1200
10x120	5	$7\frac{1}{2}$	1200
14x 72	10	—	1200
18x120	10	15	1200
18x144	10	15	1200
18x168	10	15	1200
18x 96	10	15	1200
44-in. car wheel grinder		30	900

APPENDIX NO. 4. CONDITIONS WHEN EQUIPPING OLD MACHINES WITH MOTOR DRIVE

1 When changing over from lineshaft drive to individual motor drive the question arrises whether to equip the old lineshaft-driven machines with motors or to install new motor-driven machine tools. The old machines are not as strong in construction as new tools designed for motor drive, nor are they equipped with the latest devices by means of which the time required to make adjustments can be greatly reduced. Owing to weaker construction old machines cannot be made to remove metal as rapidly as machines built with this point in view. The old machines are also more or less worn and not as accurate as new machines. A concrete example will show a method of arriving at a decision between attaching a motor to an old machine and purchasing a complete new motor-driven equipment.

2 The case taken for consideration involves the modification or exchange of a 72-in. vertical belt-driven boring mill, so as to obtain a greater output at lower cost per unit of product. This mill, the original cost of which was $3200, has been in use five years. The hourly overhead operating charge has been determined at 91 cents. The machinist receives 35 cents an hour for 54 hours per week (2808 hr. per year). The total earnings for the year from this machine amount to $4200. The operating expenses for the year are as follows:

Overhead	0.91 × 2808 =	$2555.28
Wages	0.35 × 2808 =	982.80
Total		$3538.08
Net profit $4200 − $3538 =		$662.00

3 The depreciated value of this tool on a basis of 10% reduced balance is 66% of its first cost. If a motor is installed the investment appears as follows:

Value of tool	$0.66 × 3200 =	$2112.00
Cost of motor, gears, controller, wiring, etc.	=	550.00
Total investment		$2662.00

4 The hourly overhead charge of 91 cents includes interest and depreciation at 16 cents an hour; the overhead change exclusive of interest and depreciation will therefore be 75 cents an hour. The depreciation on the new investment for the remaining five years' life of the tool will be 20% per year, making the

charge for interest and depreciation 26%. The operating cost of the old tool with motor drive is therefore,

Overhead (exclusive of interest and depreciation) \$0.75 × 2808	= \$2106.00
Interest and depreciation, 26% of \$2662	= 692.12
Wages, \$0.35 × 2808	= 982.80
	\$3780.92

Assuming 10% increased earnings, due to adoption of individual motor drive, makes the total earnings:

\$4200 + \$420 = \$4620.00

The net profit is then

\$4620 - \$3780.92 = 839.08

or 31.5% interest on the investment of \$2662.

5 The corresponding figures based on the installation of a new machine tool with individual motor drive are approximately as follows:

Cost of new tool	=	\$3400.00
Cost of motor etc.	=	270.00
		\$3670.00
Scrap value of old tool at 5%		160.00
Investment		\$3510.00
Overhead operating charge		
\$0.75 × 2808	=	\$2106.00
Wages as above		982.80
Interest and depreciation for 10 years (depreciation 10% interest 6%) 16% × \$3510	=	561.60
Total		\$3650.40

Assuming 25% increased output for the year, the total earnings become

125% × \$4200 = \$5250.00

Net profit is then \$5250 – \$3650.40 = \$1599.60

or 45.3% interest on the investment.

CONCLUSIONS

6 The above figures show that for the conditions given, approximately 14% greater return on the investment is gained by installation of a complete new tool. It is evident, therefore, that although a somewhat greater capital is required for the new installation, it is by far the better investment. It is also probable that the old machine tools would not last more than five years after the changes were made, whereas the new tools will give good service for at least double that period. Furthermore, the new machine has the added advantage of being in first class condition, thus insuring greater accuracy of workmanship and less liability to accidental delays.

APPENDIX NO. 5. OVERHEAD CHARGES AND MACHINE-HOUR RATES

The following analysis outlines a method of determining the hourly overhead charges per machine tool, which will be called the *machine-hour rates*. Overhead charges can be grouped in three main classes:

A Charges against the entire factory.

a Fixed charges: these include interest and depreciation, taxes and insurance on buildings, grounds and accessories.

TABLE 1 MACHINE HOUR RATES

Type of Machine	Charges per Hour				Depreciation	Power	Total or Mch. Hr. Rate
	Fixed	Variable	Salaries	Interest			
Vertical Boring Mills.							
40-in.-60 in.	$0.02	$0.25	$0.15	$0.05	$0.05	$0.01	$0.53
72 in.-100 in.	0.04	0.45	0.25	0.08	0.08	0.01	0.91
10 ft.-14 ft.	0.05	0.80	0.40	0.15	0.15	0.02	1.57
16 ft.-24 ft. Ext.	0.08	2.00	1.00	0.30	0.30	0.03	3.71
Average per cent of total	3%	52%	28%	8%	8%	1%	100%
Radial drills, 5 ft	$0.02	$0.30	$0.20	$0.03	$0.03	$0.01	$0.59
Radial drills, 10 ft.	0.04	0.60	0.35	0.09	0.09	0.01	1.18
Average Per Cent of Total	3%	51%	31%	7%	7%	1%	100%
Engine Lathes:							
30 in.-40 in.	$0.02	$0.25	$0.12	$0.04	$0.04	$0.01	$0.48
40 in.-60 in.	0.03	0.50	0.25	0.10	0.10	0.01	0.99
Average Per Cent of Total	3%	51%	25%	10%	10%	1%	100%
Planers:							
36 in.-56 in.	$0.04	$0.55	$0.30	$0.05	$0.05	$0.01	$1.00
7 ft.-10 ft.	0.06	1.10	0.60	0.15	0.15	0.02	2.08
12 ft.-14 ft.	0.15	2.60	1.40	0.25	0.25	0.03	4 68
Average Per Cent of Total	3%	55%	30%	5.5%	5.5%	1%	100%

b Variable charges: these include repairs and renewals on buildings and accessories, omitting all charges which can be set off directly to a particular section of the factory; charges against the store room and the tool room; defective design, material or workmanship; printing and stationery; lubricants and general manufacturing supplies.

c Salaries (not chargeable to a definite section): these include cost of superintendence (manager, superintendent, foreman); engineering and drawing; clerical force, including office boys and general laborers.

B Charges against each section of the factory.

a Fixed charges; including an equitable portion of the total factory fixed charge and interest, and depreciation on auxiliary apparatus located in the section (except machine tools).

b Variable charges: these include a portion of the variable charges as well as similar charges belonging to the section, such as repairs and renewals, storeroom and tool room charges, defective design, material and workmanship, lubricants and manufacuring supples.

c Salaries: including a portion of the total salaries as well as those belonging exclusively to the section, that is, foremen, clerks, errand boys, laborers, cranemen, etc.

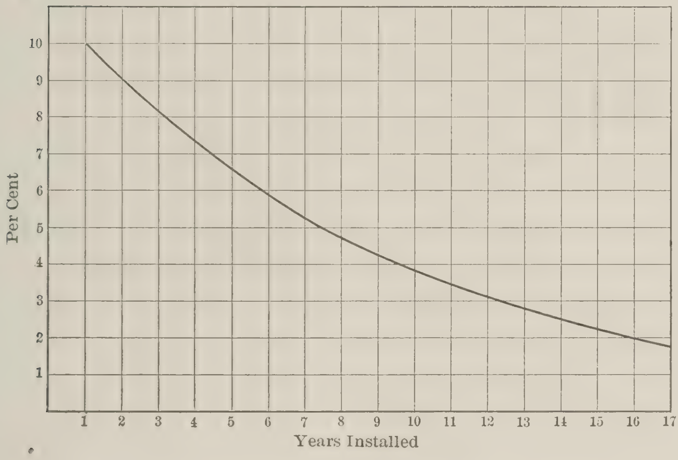


FIG. 1 DEPRECIATION AT 10%, REDUCING BALANCE

C Charges against each machine tool.

a Portion of fixed charge.

b Portion of variable charge.

c Portion of salaries charge.

d Interest on cost of tool, fairly taken at 6%.

e Depreciation of value of tool (see explanation below).

f Cost of power to operate tool, including also lighting and crane service.

DEPRECIATION OF VALUE OF MACHINE TOOLS

2 A method frequently used in calculating the depreciation in value of a machine tool is to allow 10% of a reducing balance; that is, 10% of the first cost if charged off the first year, 10% of the remaining cost, the second year, and 10% of the second remainder the third year, etc. This method is based upon the fact that the apparatus actually decreases in value year by year. Allowance for depreciation in any given year can be made easily by the aid of the curve in Fig. 1. This curve gives the percentage of the first cost corresponding each year to 10% on the reduced balance. For example, the curve shows that the depreciation on a tool that has been in service five years will be 6. 6% of the original cost. If this cost was $4500, the allowance for depreciation during the sixth year according to the 10% reducing balance method is $4500 × .066 = $297. Since this is 10% of the reduced cost, the value of the tool at the end of the fifth year is $2970.

3 Tools designed for special work will be discontinued after a comparatively limited period, and therefore, depreciate in value much more rapidly than is indicated by the foregoing method: a special allowance frequently made for such tools is generally known as *utility depreciation.*

4 Table 1 contains a summary of machine hour rates obtained by this method. It is assumed that machines have been installed six years, so that the depreciation is 6% on a basis of 10% reducing balance.

GENERAL NOTES

AMERICAN SOCIETY OF CIVIL ENGINEERS

At the meeting of the American Society of Civil Engineers, March 2, in the Society Building, 220 W. 57th Street, New York, a paper entitled, The Improved Water and Sewage Works of Columbus, O., was presented by John H. Gregory.

On March 16th two papers were presented, A Concrete Water Tower by A. Kempkey, Jun.Am.Soc.C.E., and Some Mooted Questions in Reinforced Concrete Design, by Edward Godfrey, Mem.Am.Soc.C.E.

AMERICAN INSTITUTE OF MINING ENGINEERS

The annual convention of the American Institute of Mining Engineers, in which the members of The American Society of Mechanical Engineers were invited to participate, opened on March 1 in the Carnegie Lecture Hall, Pittsburg. In the absence of Julian Kennedy, Dr. John A. Brashear, Mem.Am.Soc.M.E., gave an address of welcome. A great many interesting papers were presented during the three days which followed, including, The Development of Hindered Settling Apparatus, by Prof. R. H. Richards of the Massachusetts Institute of Technology; The Systematic Exploitation of the Pittsburg Coal Seam, by F. Z. Schellenberg of Pittsburg; A Commercial Fuel Briquette Plant, by W. H. Blauvelt of Syracuse, N. Y., Mem.Am.Soc.M. E.; The Gaseous Decomposition Products of Black Powder, by C. M. Young, Lawrence, Kan.; A New Method of Cyaniding Gold and Silver Ores, by E. Gibbon Spilsbury of New York, Mem.Am.Soc.M.E.; The Huronian as a Gold Bearing Terrane, by Dr. Robert Bell, of the Canadian Geological Survey; The Introduction of the Basic Steel Process in the United States, by Geo. W. Maynard of New York; Electric Mine Hoists, by David B. Rushmore of Schenectady, N. Y., Mem.Am.Soc.M.E.; and The Investigations of Structural Materials for Use in Federal Buildings, by E. F. Burchard of the Geological Survey. J. A. Holmes, of the U. S. Geological Survey, Mem. Am. Soc. M. E., also gave a brief paper on the work of the technological branch

at Pittsburg, which was largely explanatory of the plant and work of the survey testing station at Pittsburg. This station was later visited by the members and a series of highly interesting tests were conducted in their presence. Other excursions were also planned and carried out successfully. On the evening of March 2, Dr. D. T. Day of Washington gave a lecture on The Accumulation of Petroleum in the Earth.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The regular monthly meeting of the American Institute of Electrical Engineers was held in the auditorium of the Engineering Societies Building, New York, on Friday, March 11, 1910. This meeting was under the auspices of the Industrial Power Committee. Papers were presented as follows: Electric Mine Hoists, by D. B. Rushmore, Mem. Am.Soc.M.E., and K. A. Pauly; and Large Electric Hoisting Plants, by Wilfred Sykes.

Institute Meetings will be held, March 30-April 1, in Charlotte, N. C., and April 21, in San Francisco, Cal. A notice of papers to be read will be found in another department. The next New York Meeting will be held April 8.

CONSERVATION DISCUSSED AT THE NEW HAVEN ECONOMIC CLUB

At a dinner given by the Economic Club of New Haven, Thursday evening, February 24, 1910, The Conservation of our National Resources was presented from different viewpoints by speakers of wide reputation. Calvin W. Rice, Secretary Am.Soc.M.E., spoke on the great work now being conducted by the Government in relation to forests, lands and minerals. He was followed by Charles N. Chadwick, one of the commissioners of the board of Water supply of New York, now constructing an aqueduct from the Catskill mountains to New York City. Mr. Chadwick addressed the gathering on the importance of the conservation of water, upon which the great majority of the other resources are dependent. George W. Woodruff, of New York, former Assistant Attorney-General, emphasized the great economic importance of the preservation of resources. The last speech, by Prof. Herman H. Chapman, acting dean of the Yale Forestry School, dwelt on conservation as related to forests and said that it had been the mission of the United States forest service to actually demonstrate the true meaning of the word conservation as applied to the forests on our public lands in the West.

Among the members and guests present were Henry B. Sargent, Mem.Am.Soc.M.E., Prof. L. P. Breckenridge, Mem.Am.Soc.M.E., Dr. W. L. Phillips, Max Adler, Samuel R. Avis, Dr. Henry Spang.

INSTITUTION OF MECHANICAL ENGINEERS

The Institution of Mechanical Engineers held its annual meeting February 18, 1910, in the institution house, Storey's Gate, St. James's Park, London, S. W., and elected the following officers for the ensuing year: J. A. F. Aspinwall, President, A. T. Tannett Walker and Edward B. Ellington, Vice-Presidents.

The President called upon Dr. Glazebrook of the National Physical Laboratory to resume the discussion on the ninth report to the Alloys Research Committee on The Properties of Some Alloys of Copper, Aluminum and Manganese, presented at the previous meeting by Dr. Rosenhaim and F. C. A. H. Lantsberry. Dr. Glazebrook was followed by Sherard Cowper-Coles, H. F. Donaldson, H. L. Heathcote and Loughnan Pendred, with a closure by Dr. Rosenhaim.

INTERNATIONAL CONGRESS OF INVENTORS

The first annual convention of the International Congress of Inventors will be held in Rochester, N. Y., June 13-18, 1910. This association was established in 1906 and incorporated in 1907, with the aim of uniting the inventors of the world for the purpose of obtaining patent law reforms and protecting the interests of its members. An exhibition of patents and models will be held in connection with the convention, and will include both recent inventions and some of those of particular interest patented during the early years of the U. S. Patent Office.

AMERICAN RAILWAY ENGINEERING AND MAINTENANCE OF WAY ASSOCIATION

At the eleventh annual convention of the American Railway Engineering and Maintenance of Way Association in Congress Hall, Chicago, Ill., March 15-17, 1910, reports were presented on uniform rules, signals and interlocking; conservation of natural resources; economics of railway location; wood preservation; standard specifications for cement, masonry and buildings, and other subjects of interest.

AIR BRAKE ASSOCIATION

At the seventeenth annual convention of the Air Brake Association to be held in Indianapolis, Ind., beginning May 10, 1910, committee reports will be received on the following topics: Air Brake Instruction, Examination and Rating; Air Pump Piping, Fittings and Connections; Best Arrangement of Air Pump and Main Reservoir Capacity for 100-Car Train Service; Brake Cylinders and Connections and Recommendations for Overcoming Troubles due to Cylinder Leakage; Questions and Answers on New York Brake Equipment; Questions and Answers on Westinghouse Equipment; Recommended Air Brake Practice; Inspection and Cleaning of Triple Valves and Brake Cylinders: the Past Year's Developments in Air Brakes.

AMERICAN ELECTROCHEMICAL SOCIETY

The Spring Meeting of the American Electrochemical Society will be held in Pittsburg, Pa., on May 4-7, 1910. On Wednesday, May 4, at 2.00 p.m., it is proposed to make a visit of inspection to the technological testing plant of the U. S. Geological Survey. Other excursions will be to the Park Company's Crucible Steel Mills, Carnegie Steel Company's Dried Blast Plant at Isabella Furnace, Jones and Laughlin's Steel Works (Talbot Continuous Steel Process), Pennsylvania Lead Smelting Company, Nernst Lamp Factory, Oxy-Actylene Welding Company; with an all-day excursion, visiting the Allegheny Plate Glass Works at Glassmere, Westinghouse Electric Works at East Pittsburg, the Firth-Stirling Works at Demmler (Heroult electric steel furnace in operation), Carnegie Steel Works at Homestead (combined open-hearth electric furnace in operation).

This is the first gathering of the society at Pittsburg. A fine program and a highly interesting meeting is assured.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

At the regular monthly meeting of the Engineers' Society of Western Pennsylvania, held March 15, 1910, President E. K. Morse, in response to a request made by the Pittsburg Chamber of Commerce, appointed the following Committee to report on the question of raising the bridges over the Allegheny River at Pittsburgh: Geo. S. Davison, Mem.Am.Soc.C.E.; Julian Kennedy, Mem.Am.Inst.M.E.; F. L. O. Wadsworth, Mem.Am.Soc.M.E.; John N. Chester, Mem.Am.Soc.-

M.E.; Emil Gerber, Mem.Am.Soc.C.E. The importance of the questions involved, both engineering and financial, may be realized from the fact that there are eight bridges affected, over a river averaging 1000 feet in width. A paper on Floods in the River Seine was read by Thos. P. Roberts of the U. S. Engineer's Office, Pittsburg.

IDAHO SOCIETY OF ENGINEERS

On February 12 the Idaho Society of Engineers was organized at Boise, Idaho, with 70 charter members representing surveyors and the four leading branches of engineering. It is the outgrowth of the Idaho Civil Engineers and Surveyors Association, and has been organized mainly through the work of Gen. Darwin A. Utter, United States surveyor-general for the State, who was elected president. In addition to the work of organization, papers were read on Dam Building, Railroad Construction, Water Power, Milling Ores, Irrigation and Municipal Engineering. Meetings will be held at Boise on the second Tuesday of each month.

UNIVERSITY OF KANSAS

The new engineering buildings of the University of Kansas were formally dedicated on February 25, 1910, in the presence of some 500 visitors, including alumni, engineers from other schools, and other interested persons. The program included three addresses in the chapel in the afternoon, by Prof. F. O. Marvin, Dean of the School of Engineering, Prof. Richard C. Maclaurin, President of Massachusetts Institute of Technology, and Ernest R. Buckley, President of the American Mining Congress. The dedication ceremony itself was held a little later, in the new mechanical and electrical engineering building, and was followed by a banquet at the Robinson gymnasium in the evening.

DETROIT INDUSTRIAL EXPOSITION

The city of Detroit, Mich., is planning a great industrial exposition to be held under the auspices of the Board of Commerce, June 20-July 6, 1910. The exposition ground will be located on the Detroit River where a huge building will be erected and used in conjunction with the Wayne Pavilion. The display promises to be one of the most unique ever arranged outside of a world's fair. It is claimed that 100,000

different articles are manufactured in the 3000 shops of the city, the products ranging from pins to steamships, a variety rivaled by the outputs of few other American cities. The processes as well as the results will be shown, with the purpose of teaching the world the variety, extent and quality of the city's products. The committee in charge is composed of 275 of the leading manufacturers of Detroit.

PERSONALS

Henry A. Allen has been appointed consulting engineer for the Department of Public Works, Chicago, Ill.

F. E. Bocorselski, who has been connected with the Baush Machine Company as superintendent and designer, has resigned his position to become assistant mechanical superintendent of the American Locomotive Company, with headquarters at Richmond, Va.

Claude A. Bulkeley, chief engineer of the board of education, St. Louis, Mo., has become associated with the firm of Marks & Woodwell, New York.

I. Francis Burton, formerly assistant superintendent of the Victor Talking Machine Company, Philadelphia, Pa., has been appointed superintendent of the company.

Charles F. Dixon has become connected with the engineering department of the New England Engineering Company, New Haven, Conn.

John M. Ewen has resigned his position as harbor commissioner of Chicago, Ill.

E. S. Farwell, consulting engineer, of New York, has become connected with the Yellow Pine Paper Mill Company, Orange, Tex., as general manager.

M. P. Fillingham, consulting engineer, New York, has assumed charge of the Eastern interests of the Fawcus Machine Company, of Philadelphia, Pa.

Francis L. Gilman, formerly associated with the American Telephone and Telegraph Company, New York, has become general manager of the Missouri and Kansas Telephone Company, Kansas City, Mo.

George P. Gilmore, recently local engineer of the American Thread Company, Fall River, Mass., has opened a power and equipment engineering office in the same city.

B. S. Hughes has severed his direct connection with the Champion Coated Paper Company, Hamilton, O., and the Champion Fibre Company, Canton, N. C., to engage in general engineering practice, with offices in Cincinnati, O.

F. W. Jackson, formerly district manager for the Harrisburg Foundry and Machine Works, Baltimore, Md., has been transferred to the managership of the company's business at Chicago.

Walter C. Kerr has been elected third vice-president of the Merchants' Association of New York.

Alfred H. Knight has become connected with the Packard Motor Car Com-

pany, Detroit, Mich. as carriage chassis engineer. He was until recently assistant professor of mechanical engineering at the University of Michigan, Ann Arbor, Mich.

F. E. Matthews, consulting refrigerating engineer, New York, has become assistant manager of the cold storage insulation department of H. W. Johns-Manville Co., New York.

Geo. R. Murray has been appointed president of the Murray Stone Co., successors to the Maxwell-Rolf Stone Company.

John C. Parker, electrical engineer for the Rochester Railway and Light Company, has been appointed non-resident lecturer in electric energy transmission at the University of Michigan, Ann Arbor, Mich.

W. P. Pressinger, formerly identified with the W. P. Pressinger Company, New York, has been appointed vice-president and manager of sales of the Keller Manufacturing Company, Philadelphia, Pa.

Paul S. Rattle, mechanical engineer of B. M. Osbun Co., Chicago, Ill., has become associated with the sales organization of the Hicks Locomotive and Car Works, Chicago.

Robert W. Rogers, formerly identified with the Erie Railroad, Meadville, Pa., has entered the service of the C. A. Stickney Co., St. Paul, Minn., as mechanical engineer.

Clement F. Smith, recently associated with the Westinghouse Air Brake Company, Wilmerding, Pa., has opened an office in Cleveland, O.

Ephraim Smith, who has been the New England sales manager of the Colonial Steel Company since its organization in 1901, has resigned his position on account of ill health.

Roy B. Smith has become inspector of the Pennsylvania Lines, West, Columbus, O. Until recently he was foreman of motive power and equipment of the C. L. & N. Railway, Cincinnati, O.

B. V. Swenson has become connected with Barron G. Collier. Inc., New York. He was formerly secretary and treasurer of the American Street and International Railway Association, New York.

Cary D. Terrell, formerly assistant manager of sales of the Pressed Steel Car Company, St. Louis, Mo., has become sales agent of the American Car and Foundry Company, St. Louis, Mo.

Henry R. Towne has been elected president of the Merchants' Association of New York.

Theron H. Tracy, president of the Tracy-Devereaux Co., Los Angeles, Cal. has been appointed president of the Durostone Company of America, San Diego Cal.

A. W. Waern has become associated with Jos. H. Wallace & Co., New York.

He was formerly engineer of the machinery department of the Bethlehem Steel Company, South Bethlehem, Pa.

Prof. Ira H. Woolson, adjunct professor of civil engineering, Columbia University, in charge of fire tests of building materials, has resigned to become consulting engineer for the National Board of Fire Underwriters.

Roydon V. Wright, for several years managing editor of the American Engineer and Railroad Journal, has become a member of the editorial staff of the Railway Age Gazette, with direct supervision over the mechanical department

CURRENT BOOKS

THE ECONOMY FACTOR IN STEAM-POWER PLANTS. By George W. Hawkins. *Hill Pub. Co., New York, 1908.* Cloth, 8vo., ix+ 133 pp., illustrated. Price, $3 net.

Contents: Introduction; Part I, Individual Apparatus: Boilers, Engines, Electrical Generators, Condensing Apparatus, Feed-Pumps, Oil-Pumps, Oil Burners, Radiation, Leakage, Feed-Water Heaters, Fuel Economizers; Part II, The Factor of Evaporation; Part III, Complete Plant Economy (Full Rated Load): Introductory, Non-Condensing Plants, Surface-Condensing Plants, Jet-Condensing Plants, Pumping Plants, Examples; Part IV, Complete Plant Economy (Variable Load): Phases of the Problem, Method of Solution; Conclusion.

THE GAS TURBINE. Progress in the Design and Construction of Turbines Operated by Gases of Combustion. By Henry Harrison Suplee, B.Sc. *J. B. Lippincott Co., Philadelphia, 1910.* Cloth, 8vo., 262 pp., with diagrams. Price, $3.

Contents: Introduction; Historical; Discussion before the Institution of Mechanical Engineers; Discussion before the Society of Civil Engineers of France; Actual Behavior of Gases in Nozzles; Practical Work of Armengaud and Lemale; General Conclusion.

HEAT ENERGY AND FUELS. Pyrometry, Combustion, Analysis of Fuels and Manufacture of Charcoal, Coke and Fuel Gases. By Hanns v. Jüptner. Translated by Oskar Nagel, Ph.D. *New York, McGraw Pub. Co.,1908.* Cloth, 8vo., 306 pp., illustrated. Price, $3.

Contents: Introduction: General Remarks, Forms of Energy. Vol. I. Heat Energy and Fuels. Part I. Heat Measurement, Combustion and Fuels: The Measurement of High Temperatures (Pyrometry); Pyrometry, Optical Methods of Measuring Temperatures; Combustion Heat and its Determination; Direct Methods for Determining the Combustion Heat; Incomplete Combustion; Combustion Temperature; Fuels (in general); Wood; Fossil Solid Fuels (in general); Peat; Brown Coal (Lignite); Bituminous and Anthracite Coals; Artificial Solid Fuels; Charcoal; Peat-Coal, Coke and Briquettes; Coking Apparatus; Liquid Fuels; Gaseous Fuels; Producer Gas; Water Gas; Dowson Gas, Blast Furnace Gas and Regenerated Combustion Gases; Apparatus for the Production of Fuel Gases.

THE RESISTANCE AND PROPULSION OF SHIPS. By William F. Durand. Second edition, thoroughly revised. *New York, John Wiley & Sons, 1909.* Cloth, 8vo., vii + 427 pp. Price, $5.

Contents: Resistance; Propulsion; Reaction between Ship and Propeller; Propeller Design; Powering Ships; Trial Trips.

THE MODERN GAS ENGINE AND THE GAS PRODUCER. By A. M. Levin. First edition. *New York, John Wiley & Sons, 1910.* Cloth, 8vo., 16 + 485 pp., illustrated. Price, $4.

Contents: Introduction to Thermodynamics; Design Constants and Formulas; Theoretical Analyses of the Gas-Engine Cycles; Power, Size and Speed of Gas-Engines; Fuels, Combustion; Gas-Engine Fuels—The Proportioning of Mixtures and Relation of these to the Size of the Engine; Alcohol Fuels; Features of the Practical Gas-Engine Cycle; The Fly-Wheel; The Crank Shaft; Engines Details; Governing; Engine Auxiliaries; Various Engine Types; Producer-Gas and Gas-Producers; Appendix.

FULFILMENT OF THREE REMARKABLE PROPHECIES IN THE HISTORY OF THE GREAT EMPIRE STATE, Relating to the Development of Steamboat Navigation and Railroad Transportation, 1808–1908. By Henry Whittemore. Cloth, 8vo., 80 pp.

Contents: Early Experiments in Steamboat Navigation; James Rumsey's Claim to the Discovery of Steamboat Navigation; Claims of Nathan Read, Nicholas Roosevelt, Capt. Samuel Morey and Elijah

Ormsbee; Inventions of Col. John Stevens and Robert L. Stevens in Steam Navigation; Robert Fulton–His Successful Efforts in the Development of Steam Navigation with the Assistance of Robert L. Livingston; Improvements in Steamboats and Increased Facilities for Steam Navigation on the Hudson River; Rivalry between Steamboat Companies—Steamboat Disasters—Great Improvement in Steamboat Construction; Railroad Transportation.

AUTOMOBILES. By Hugo Diemer, M.E. *Chicago, American School of Correspondence, 1909.* Cloth, 192 pp., illustrated. Price, $1.50.

Contents: Component Parts of a Motor-Car; Power Plant of a Gasolene Car; Controlling Mechanism and Transmission; Care and Operation of Motor-Cars; Selection and Classification of Motor-Cars; Index.

TABLES AND DIAGRAMS OF THE THERMAL PROPERTIES OF SATURATEDAND SUPERHEATED STEAM. By Lionel S. Marks and Harvey N. Davis. *New York, Longmans, Green & Co., 1909.* Cloth, 8vo., 106 pp.,illustrated. Price, $1, net.

Contents: Part I: Tables and Diagrams; Part II: The Use of the Diagrams; Part III: Discussion of Sources.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. List of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

American Mining Congress. Monthly bulletin. Vol. 13, no. 2. *Denver, 1910.*

American Water Works Association. Proceedings, 1909. *Baltimore, 1909.* (Gift of American Water Works Association.)

Approximate Cost of Mill Buildings. By C. R. Main. *Waltham, 1910.*

Automobiles. By Hugo Diemer. *Chicago, 1909.* (Gift of author.)

Beiträge zür Geschichte der Technik und Industrie. Vol. 1, 1909. *Berlin, Springer, 1909.* (Gift of Verein deutscher Ingenieure.)

Broadening the Field of the Marine Steam Turbine: the Problem and its Solution. The Melville and Macalpine Reduction Gear. *Pittsburg, 1909.*

Carnegie Institution of Washington, Department of Terrestrial Magnetism. Annual report of the director, 1909. (Gift of the Department.)

Civil Engineer's Pocket-book. Ed. 19. By J. C. Trautwine. *New York, J. Wiley & Sons, 1909.* (Gift of J. C. Trautwine, Jr., and J. C. Trautwine, 3d.)

Control of Flies and Other Household Insects. Bulletin 136, N. Y. State Museum. By E. P. Felt. *Albany, 1910.*

Design and Construction of Internal-Combustion Engines. By Hugo Guldner. *New York, 1910.*

Deutsch-Amerikanischen Techniker-Verbandes. Verbands-Statuten, 1910. *New York, 1910.*

Dictionnaire de la Langue Française. Vol. 1–4 and supplement. *Paris, 1884–1885.*

Economy Factor in Steam-Power Plants. By G. W. Hawkins. *New York, Hill Publishing Co., 1908.*

Elements of Machine Design. Part 1, General principles, strength of materials, etc. *London, 1909.*

Elevator Service. By R. P. Bolton. *New York, 1908.*

Energy: Work, Heat and Transformations. By S. A. Reeve. *New York, McGraw-Hill Book Company, 1909.*

ENGINEERS' AND FIREMEN'S LICENSE LAW. BOILER INSPECTION LAW. Rules Formulated by the Board of Boiler Rules, Commonwealth of Massachusetts. *Boston, 1909.* (Gift of John A. Stevens.)

GAS ENGINE. By C. P. Poole. *New York, Hill Publishing Co., 1909.* (Gift of author.)

DIE GASMASCHINE. Ed. 5. By R. Schöttler. *Berlin, 1909.*

GAS TURBINE. By H. H. Suplee. *Philadelphia, 1910.* (Gift of author.)

GROSSGASMASCHINENBAU IN AMERIKA. By Dr. Rieppel, Jr. (Reprint Zeitschrift Vereines deutscher Ingenieure, 1909.) (Gift of author.)

HANDBOOK OF SMALL TOOLS. By Erik Oberg. *New York, 1908.*

HEAT ENERGY AND FUELS. By Hanns v. Juptner. *New York, McGraw Publishing Co., 1908.*

HENLEY'S ENCYCLOPÆDIA OF PRACTICAL ENGINEERING AND ALLIED TRADES. Vol. 5 (Spe-Z). *New York, N. W. Henley Publishing Co., 1909.*

HYDRAULIC ELEVATORS. By Wm. Baxter, Jr. *Chicago, 1905.*

ILLUSTRATED TECHNICAL DICTIONARY. Vol. 5, Railway construction and operation; Vol. 6, Railway rolling stock. *New York, 1909.*

LARGE GAS ENGINES. By P. R. Allen. (Reprinted from Cassier's Magazine, July-September 1909.) (Gift of Cassier's Magazine.)

LINSEED OIL AND OTHER SEED OILS. By W. D. Ennis. *New York, D. Van Nostrand Co., 1909.*

LOSSES OFF TRANSMISSION LINES, DUE TO BRUSH DISCHARGE, WITH SPECIAL REFERENCE TO THE CASE OF DIRECT CURRENT. (Institution of Electrical Engineers, 1909.) (Gift of Calvin W. Rice.)

MILLWRIGHTING. By J. F. Hobart. *New York, 1909.*

MODERN ELECTRIC TIME SERVICE. By F. H. Jones. (Institution of Electrical Engineers, 1909.) (Gift of Calvin W. Rice.)

MODERN GAS-ENGINE AND THE GAS-PRODUCER. By A. M. Levin. *New York, J. Wiley & Sons, 1910.*

MOTOR TRACTION. Vol. 9, no. 421-date. *London, 1909-date.*

NEW YORK CITY, DEPARTMENT OF BRIDGES. Report on Manhattan Bridge. By Ralph Modjeski. 1909. *New York, 1909.* (Gift of the Department.)

POSTULADOS DE LAS CLASES OBRERAS Y DE LOS DESCALIDOS Y PROLETARIOS, A PRESENCIA DE LA CIENCIA SOCIAL Y, EN ESPECIAL, DE LA ECONOMIA POLITICA. Vol. 2. *Santiago de Chile, 1909.* (Gift of Secretary-General, Fourth Scientific Congress [First Pan-American].

PRACTICAL COLD STORAGE. By Madison Cooper. *Chicago, 1905.*

PRINCETON UNIVERSITY. Directory of Living Graduates and Former Students, 1908. *Princeton, 1908.*

PRODUCER-GAS-FIRED FURNACES. By Oscar Nagel. *New York, 1909.*

Resistance and Propulsion of Ships. Ed. 2. By W. F. Durand. *New York J. Wiley & Sons, 1909.*

Smithsonian Institution. Annual report. 1908. *Washington, 1909.*

Société des Ingenieurs Civils de France. Inauguration du Nouvel Hotel, January 1897. *Paris, 1897.*

Steam Power Plant Piping Systems. By W. L. Norris. *New York, 1909.*

Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam. By L. S. Marks and H. N. Davis. *New York-London, Longmans, Green & Co., 1909.* (Gift of author.)

Theory and Practice of Modern Framed Structures. Part 1, Stresses in simple structures. Ed. 9. By J. B. Johnson, C. W. Bryan and F. E., Turneaure. *New York, 1910.*

Types and Details of Bridge Construction. Parts 1–3. By F. W. Skinner. *New York, 1904, 1906, 1908.*

U. S. Library of Congress. Duplicate periodicals and serials available for exchange January 1910. *Washington, 1910.*
———Want list miscellaneous publications 1909. *Washington, 1909.*

U. S. Library of Congress. Report of Librarian, 1909. *Washington, 1909.*
———Publications issued since 1897. *Washington, 1910.*

University of Pennsylvania. Catalogue, 1909–1910. *Philadelphia, 1910.*
Verein deutscher Ingenieure. Zur feier des 50 jährigen bestehens des vereines. 1856–1906.

———Berliner bezirtsverein deutscher Ingenieure. 1856–1906. *Berlin, 1900.*

Watuppa Water Board. 36th Annual Report. *1910.*

EXCHANGES

Junior Institution of Engineers. Journal and Record of Transactions. Vol. 19. *London, 1909.*

Liverpool Engineering Society. Transactions. *Liverpool, 1909.*

Railway Signal Association. List of Members, 1910. *Bethlehem, 1910.*

Sächsischer Dampfkessel Revisions Verein, Chemnitz. Ingenieur-Bericht, 1909. *Chemnitz.*

TRADE CATALOGUES

American Spiral Pipe Works, *Chicago, Ill.* Spiral riveted pipe, forged steel pipe flanges, hydraulic and exhaust steam supplies, 20 pp.

Asbestos Protected Metal Co., *Canton, Mass.* Asbestos protected metal for roofing, siding, ceiling, and interior finish, 8 pp.

ECONOMY DRAWING TABLE CO., *Toledo, O.* Drawing tables, sectional filing cases and specials in this line, for engineers, architects, contractors, manual training schools, etc., 48 pp.

GENERAL ELECTRIC CO., *Schenectady, N. Y.* Building lighting with general electric tungsten and tantalum lamps, 16 pp. Price list No. 5211, G. E. tantalum incandescent lamps, 5 pp.; November 1909, Index to bulletins published, 9 pp.; Bulletin No. 4703A, Variable release air brake equipment, 11 pp.; Bulletin No. 4714, Railway signal voltammeter, type S, 3 pp.; Bulletin No. 4715, G. E. 210 Railway motor, 16 pp.

HAGAN GAS ENGINE & MFG. CO., *Winchester, Ky.* Catalogue C, 2 to 100 h.p. Hagan gas and gasolene engines, 39 pp.

JEFFREY MFG. CO., *Columbus, O.* Booklet No 33, Jeffrey wire cable conveyors, 24 pp.; Booklet No. 34, Jeffrey standard elevator buckets, 24 pp.

LAMSON CONSOLIDATED STORE SERVICE CO., *Boston, Mass.* Two-wire cord-propulsion parcel carriers for stores, 8 pp.

MANUFACTURING EQUIPMENT AND ENGINEERING CO., *Boston, Mass.* All-metal, sanitary, and fireproof equipment for factories, foundries, offices, hospitals, etc., 32 pp.

NATIONAL VACUUM HEATING CO., *Marshalltown, Iowa.* Dunham vacuo-vapor system of heating, 40 pp.

NORTHWESTERN EXPANDED METAL CO., *Chicago, Ill.* Reinforcing for sewers, tanks, and walls, 16 pp.

OHIO BRASS CO., *Mansfield, O.* Bulletin of electric railway and mine haulage material, 24 pp.

REMINGTON TYPEWRITER CO., *New York, N. Y.* Remington Notes, vol. 2, No. 2, containing notes of interest to users of the Remington typewriter, 16 pp.

FRANCIS H. RICHARDS, *New York, N. Y.* Useful information concerning patents and inventions, 38 pp.

RUSSEL WHEEL AND FOUNDRY CO., *Detroit, Mich.* Views of Russel skidding and loading machines in operation, 40 pp.; catalogue of different styles and patterns of Russel cars for handling logs, lumber, etc., 46 pp.

JOSEPH T. RYERSON & SONS, *Chicago, Ill.* March, 1910. Ryerson Monthly Journal and Stock List of iron and steel supplies, 144 pp.

CHAS A. STICKNEY CO., *St. Paul. Minn.*, Bulletin No. 1137, The Stickney oil engine and 57 points in which it excels other engines, 16 pp.

UNDERFEED STOKER CO. OF AMERICA. *Chicago, Ill.* Publicity Magazine, February 1910, devoted to the interests of the Jones mechanical stoker, 15 pp.

WAGNER ELECTRIC MFG. CO., *St Louis, Mo.* Bulletin 89—Type BW polyphase induction motor, 8 pp.

WARNER & SWASEY CO., *Cleveland, O.* Warner & Swasey prism terrestrial telescope, 3 pp.

UNITED ENGINEERING SOCIETY

GIFT OF E. E. OLCOTT

ALBANY INSTITUTE. Transactions. Vol. 5. *Albany, 1867.*

NEW YORK STATE. Adjutant General. Annual report. Vol. 3. *Albany, 1868.*

NEW YORK STATE ENGINEER AND SURVEYOR. Annual report on canals. 1862, 1864. *Albany, 1863, 1865.*

———Annual report on railroads. 1862, 1865. *Albany, 1863, 1866.*

NEW YORK STATE RAILROAD COMMISSIONERS. Report, Vol. 1, 1885. *Albany, 1886.*

SWEET, S. H. Documentary sketch of New York State canals. *Albany, 1863.*

REGISTER TILL PATENT MEDDILADE AF KUNGL. PATENTBYRAN. 1885–1908 and supplement to 1905. *Stockholm, 1890–1909.*

TRADE CATALOGUES

GIROD FURNACES, *Ugine, Savoie.* Steels made by the Girod process, 16 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

015 Sales manager for improved type of heavy-duty gas and gasoline engines, up to 25 h.p., for industrial and farm purposes. Applicant should state experience in similar capacity and what results he could agree to produce.

016 Large blast-furnace plant in the South wants at once draftsman between twenty-five and thirty years old, technically educated, with sufficient breadth to do testing and various work about the plant. Climate agreeable and healthful. Exceptional opportunity for the right man.

017 Wanted, by a large iron and steel company, superintendent of shops; include pattern, foundry, blacksmith, pipe and machine shops; combined force of about 500 men. Prefer technically educated man. Must be an organizer, familiar with modern methods and able to hold production costs on reasonable basis. None but high class man need apply. Salary $3600 per annum.

018 Wanted: thorough practical and theoretical man, to take charge of the production and the development of a concern located in the Middle West, manufacturers of injectors, valves, etc. Must be familiar with railroad operating conditions and thoroughly up-to-date in brass foundry and machine shop practices.

019 Instructor in mechanical drawing and machine design in a technical school near New York. Previous teaching experience desirable but not essential. Salary about $1500 with good opportunity for advancement.

020 Assistant engineering editor on prominent trade journal; excellent opportunity for rapid advancement. State experience fully; communications confidential.

021 Opportunity for engineer with business experience to acquire interest in business of manufacture of all classes of hydraulic machinery, steam hammers, etc. Location Pennsylvania.

MEN AVAILABLE

41 Lawyer-engineer, desires position as salesman. Age 44; member United States Supreme Court; legal and technical education, twenty-five years successful experience combined with sales, steam, electrical and gas driven power plants, water and gas works, chemicals, every description of machinery and its product; specialty, automobile salesman. Extensive acquaintance throughout the United States.

42 University graduate in mechanical engineering, a student member, located on the Pacific Coast, desires position with eastern gas engine company; at present designing medium and large liquid fuel engines, stationary and marine; testing and machine shop experience.

43 Junior member, graduate mechanical and electrical engineer, Mass. Inst. of Technology; experience in erection and operation of electric power plants; has served time in large railroad shops and understands shop methods thoroughly. Location immaterial.

44 Mechanical engineer, specialized in manufacturing, thoroughly competent to take responsible position. As superintendent and manager has successful practical experience in foundry and machine shop; gray iron and brass mixtures by analysis, machine molding, interchangeable machine work, systematizing, cost-keeping, piece work, etc.; knows how to equip plant and organize men to secure large output and low costs.

45 Cornell graduate, married, nine years' experience with engineers, contractors and industrial companies, in drafting room, office and on construction; desires to make a change.

46 Mechanical engineer, Member, technical graduate; broad practical knowledge of engineering, good systematizer, especially able as a producer, severa- years' experience in engineering work, in charge of large engineering departl ments; desires position with first-class firm as chief engineer, or similar position.

47 Man with seventeen years' experience in office and shop of manufacturing concern, general experience in this line and executive work; would like to meet some responsible concern in New York or vicinity who want salesman or competent office manager.

48 Engineer, thirty years old, technical graduate, desires position preferably in Chicago, as factory manager of a small but growing plant. Four years' shop and drafting experience, four years installing cost and shop systems, experienced in laying out and constructing power plants and industrial works.

49 Member, with over twenty years' practical experience in designing, superintending and managing work in shop and field, desires position, preferably near Philadelphia.

50 Engineer, wants the New York agency for the best automobile delivery wagon and automobile truck in the United States.

51 Member, now general mechanical superintendent, desires change. Ten years' experience as mechanical engineer and superintendent, general transmission machinery, gas and Corliss engines. Thoroughly posted on rapid foundry, machine shop production and up-to-date appliances and methods.

52 Associate member, twenty-eight years of age, graduate marine and mechanical engineer, wishes to locate with some good, growing manufacturing concern in capacity of chief engineer, assistant chief engineer, chief draftsman or similar position, where the services of a mechanical engineer with an excellent theoretical, as well as practical training in the gas engine, producer, steam engine, power transmission and general engineering lines will be appreciated. Best references as to ability and character.

53 Associate, graduate mechanical engineer, fourteen years' experience in general engineering work, including machine shop work, testing, power plant design, construction and operation. Five years in electric railway work, involving civil, mechanical and electrical engineering; recently completed the remodeling of an electric railway and lighting power plant, now completing the construction of a large electric power plant. Good executive ability, experience in office methods, correspondence, etc. Wishes an executive position involving responsibility. Salary $2500.

54 Junior, technical graduate, at present mechanical engineer and assistant to manager of plant building high grade boilers. Experienced boiler designer. Desires similar position, or as assistant superintendent in large plant.

55 Junior member, age thirty-one, desires to make a change. Several years designing, testing and installing steam turbines, steam engines, condensers, etc.; varied experience in engineering lines, and electrical work. Desires position with large industrial corporation, or in office of consulting or contracting engineer.

56 Graduate mechanical engineer. Harvard University, S.B. and M.M.E., twenty-five years of age, experienced in the organization and management of work shops, would like to obtain position with establishment manufacturing standard line of goods, or with consulting engineer engaged in workshop organization and management.

57 Member, desires position as mechanical engineer with concern developing new inventions; competent in designing, perfecting and simplifying mechanisms.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ALDEN, Herbert W. (1908), Ch. Engr., Timken-Detroit Axle Co., Clark Ave., Detroit, Mich.

BORDEN, Wm. H. (Junior, 1905), Goldsboro, N. C., and *for mail*, 713 Seventh Ave., N., Seattle, Wash.

BULKELEY, Claude A. (1909), Cons. Mech. and Elec. Engr., 511 Terminal Bldg., 41st St. and Park Ave., New York, N. Y.

BURTON, Isaac Francis (1908), Supt., Victor Talking Mch. Co., and *for mail*, 5219 Walnut St., Philadelphia, Pa.

BUSH, Harold Montford (1894; 1905), Cons. Engr., 69 N. Fourth St., Columbus, O.

CAMPBELL, Jeremiah (Associate, 1896), 2 New St., East Boston, Mass.

CHESS, Harvey B., Jr. (Junior, 1909), Secy. and Wks Mgr., Consolidated Expanded Metal Cos., Rankin, and *for mail*, 814 Aiken Ave., Pittsburg, Pa.

COFFIN, Howard E. (1907), V. P., Hudson Motor Car Co., and 434 Cadillac Ave., Detroit, Mich.

DIXON, Charles F. (Junior, 1903), Engrg. Dept., New England Engrg. Co., 113 Church St., and *for mail*, 172 Ellsworth Ave., New Haven, Conn.

DOUGLASS, Wm. M. (1884), 306 Seventh Ave., Bethlehem, Pa.

FARWELL, E. S. (1899), Genl. Mgr., Yellow Pine Paper Mill Co., Orange, Tex.

FLEMING, Wills M. (1905; 1909), Ch. Draftsman, Deane Steam Pump Co., and *for mail*, 370 Maple St., Holyoke, Mass.

GILMAN, Francis L. (1908), Genl. Mgr., Missouri & Kansas Telephone Co., Kansas City, Mo.

GILMORE, George Parley (1909), Power and Equip. Engr., First Natl. Bank Bldg., and *for mail*, 109 Barre St., Fall River, Mass.

HILL, Robert J. (Associate, 1904), 810 Marquette Bldg., Chicago, and 816 Sheridan Road, Wilmette, Ill.

HORTON, William H. (Junior, 1904), 7001 S. Park Ave., Chicago, Ill.

HUGHES, Burton Shelley (1908), Cons. Engr., 1014 Commercial Tribune Bldg., Cincinnati, O.

JACKSON, F. W. (1909), Dist. Mgr., Harrisburg Fdy. & Mch. Wks., 950 Marquette Bldg., Chicago, Ill.

KEITH, Thomas M. (Junior, 1905), Robins Conveying Belt Co., Park Row Bldg., New York, N. Y.

KNIGHT, Alfred H. (1909), Carriage Chassis Engr., Packard Motor Car Co., and *for mail*, 185 Seward Ave., Detroit, Mich.

LEE, Ralph A. (Junior, 1909), Asst. Bldg. Supt., with Walter Kidde, 140 Cedar St., New York, and *for mail*, 578 75th St., Brooklyn, N. Y.

LEE, Robert E. (Junior, 1907), 129 Chestnut St., Rutherford, N. J.

MATTHEWS, Fred Elwood (Junior, 1904), Asst. Mgr., Cold Storage Insulation Dept., H. W. Johns-Manville Co., 100 William St., New York, N. Y.

MEINHOLTZ, Herman Chas. (1909), V. P. and Supt., Heine Safety Boiler Co., 2449 E. Marcus Ave., and *for mail*, 4812 Greer Ave., St. Louis Mo.

MILLETT, Kenneth B. (Junior, 1908), Factory Supt., Protal Co., Bridgeport, Conn.

MINCK, Peter (Junior, 1909), Mech. Engr., with Edwin Burhorn, 71 Wall St., New York, N. Y., and *for mail*, 112 Gardner St., Union Hill, N. J.

MURRAY, Geo. R. (1903), Pres., Murray Stone Co., 914 Williamson Bldg., Cleveland, O.

MURRIE, John L. (Junior, 1905), Mech. Engr., Pub. Service Com., First Dist., Tribune Bldg., and *for mail*, 551 W. 178th St., New York, N. Y.

ORD, Henry C. (1905), Genl. Elec. Co., and *for mail*, 3 Eastern Ave., Lynn, Mass.

POULTNEY, John Livingston (1908), Contr. Engr., Land Title Bldg., Philadelphia, Pa.

POWELL, E. Burnley (Junior, 1904), Stone & Webster Engrg. Corp., 147 Milk St., Boston, Mass., and *for mail*, Houghton County Elec. Light Co., Houghton, Mich.

RATTLE, Paul S. (Junior, 1908), Sales Organization, Hicks Loco. & Car Wks., Chicago, and 459 Oak Park Ave., Oak Park, Ill.

REID, John Simpson (1898), Instr. Mech. Drawing and Design, Armour Inst. Tech., and 43 W. 33d St., Flat C, Chicago, Ill.

RICE, Alva C. (1890), Cons., Hyd. and Mech. Engr., 5 Oberlin St., Worcester, Mass.

ROUVEL, George W. (1907), Genl. Supt., Standard Portland Cement Co., Napa Junction, and Napa, Cal.

SERGEANT, Chas. H. (1895), 511 W. 134th St., New York, N. Y.

SMITH, Roy B. (Junior, 1905), Inspr., Pa. Lines West, and *for mail*, 106 S. Champion Ave., Columbus, O.

TALCOTT, Robt. Barnard (1907), Inspr. Mech. and Elec. Engrg., U. S. Post Office Bldg., Denver, Colo.

TAYLOR, Percy B. (1909), Cons. Engr., 196 Market St., Newark, N. J.

TERRELL, Cary D. (Junior, 1901), Sales Agt., Am. Car & Fdy. Co., 915 Olive St., St. Louis, Mo.

TRACY, Theron H. (1902), Pres., Durostone Co. of America, San Diego, Cal.

UNGER, John S. (1886), Cons. Engr., 1412 N. Y. Life Bldg., and *for mail*, 3344 Evanston Ave., Chicago, Ill.

WAERN, A. W. (1908), Cons. Engr., Jos. H. Wallace & Co., Temple Court Bldg., New York, N. Y.

WALSH, Thomas J. (Junior, 1906), Stone & Webster Engrg. Corp., Boston, Mass., and *for mail*, Houghton County Elec. Light Co., Houghton, Mich.

WILDER, Clifton W. (1907), Asst. Elec. Engr., Pub. Service Com., 154 Nassau St., New York, N. Y.

WRIGHT, Roydon V. (1907), Supv. Mech. Dept., Railway Age Gazette, New York, N. Y., and *for mail*, 285 N. 20th St., East Orange, N. J.

NEW MEMBERS

BLANCHARD, Henry W. (1909), Mgr., Austral Iron Wks., E. W. Tarry Co., Ltd., and *for mail*, P. O. Box 1098, Johannesburg, Transvaal, South Africa.

RUCKER, B. Parks (1909), Elec. and Mech. Engr., Trust Bldg., Charlotte, N. C.

STODDARD, Elliott J. (1909), Parker & Burton, 603 Moffat Bldg., Detroit, Mich.

PROMOTIONS

HVID, Rasmus M. (1907; Associate, 1909), Am.Soc.M.E., 29 W. 39th St., New York, N. Y.

THURN, Theodore (1904; 1909), Engr., Genl. Elec. Co., 23 Water St., Yokohoma, Japan.

DEATHS

GOODALE, A. M., December 17, 1909.

GAS POWER SECTION

CHANGES OF ADDRESS

FLEMING, W. M. (1909), Mem.Am.Soc.M.E.
MATTHEWS, Fred E. (1908), Mem.Am.Soc.M.E.
MYERS, Theodore B. (Affiliate, 1909), 981 Bulls Ferry Rd., Woodcliff-on-Hudson, N. J.
SERGEANT, Chas. H. (1908), Mem.Am.Soc.M.E.
UNGER, John S. (1909), Mem.Am.Soc.M.E.
WILDER, Clifton W. (1908), Mem.Am.Soc.M.E.

NEW MEMBERS

BIGELOW, Lucius S. (Affiliate, 1910), Pres., Light Pub. Co., 106 Fulton St., and Pres., Periodicals Pub. Co., New York, N. Y.
HARRIS, William J., Jr. (Affiliate, 1910), Junior Engr., Tech. Branch, U. S. Geolog. Survey, Washington, D. C.
HOBART, Frank G. (1910), Mem.Am.Soc.M.E.
IRWIN, Arthur Charles (Affiliate, 1910), Erecting Engr., Tait Producer Co., New York, and *for mail*, 33 E. Smith Ave., Corona, L.I., N. Y.
KENNEY, Lewis H. (1910), Mem.Am.Soc.M.E.
MYERS, David M. (1910), Mem.Am.Soc.M.E.
RANDALL, Dwight T. (1910), Mem.Am.Soc.M.E.
SMITH, Earl B. (Affiliate, 1910), Asst. Prof. Expl. Mech. Engrg., Drexel Inst., Philadelphia, Pa.

STUDENT BRANCHES

CHANGES OF ADDRESS

CHU, P. F. (Student, 1909), 31 Inman St., Cambridge, Mass.
CUMPSTON, E. H., Jr. (Student, 1909), 2252 Washington Ave., Cincinnati, O.
GREEN, J. B. (Student, 1909), 4734 Kimbark Ave., Chicago, Ill.
HARKNESS, C. L. (Student, 1910), Association House, Champaign, Ill.
HOLLENBERGER, Theo. J. (Student, 1909), 63 N. Adolph Ave., Akron, O.
ILLMER, G. M. (Student, 1909), 2739 Calvert St., Baltimore, Md.
KUPPATRICK, H. J. (Student, 1909), Roseville, Ill.
LAWRENCE, J. H. (Student, 1909), 3120 Broadway, New York, N. Y.
NYLAND, Evert (Student, 1909), 1428 N. Bouvie St., Philadelphia, Pa.
PETERSON, A. G. (Student, 1909), Lodi, N. Y.
STEINBECK, C. E. (Student, 1909), 1029 N. Hunter St., Stockton, Cal.
STEWART, H. M. (Student, 1910), 2358 Ohio Ave., Cincinnati, O.

NEW MEMBERS

ARMOUR INSTITUTE OF TECHNOLOGY

CUMMINGS, G. F. (Student, 1910), 3360 Prairie Ave., Chicago, Ill.
HATMAN, J. G. (Student, 1910), 3653 Calumet Ave., Chicago, Ill.

BROOKLYN POLYTECHNIC INSTITUTE

BURKE, T. F. (Student, 1910), 125 W. 111th St., New York, N. Y.
HELWIG, Arthur (Student, 1910), 10th Ave. and 70th St., Brooklyn, N. Y.

CORNELL UNIVERSITY

DEXTER, R. L. (Student, 1910), 603 E. Seneca St., Ithaca, N. Y.
KONSTANKEWICZ, M. (Student, 1910), 208 Williams St., Ithaca, N. Y.
LEHMAN, M. G. (Student, 1910), Barnes Hall, Ithaca, N. Y.
MATTHAI, A. M. (Student, 1910), 810 University Ave., Ithaca, N. Y.
ROOS, D. G. (Student, 1910), 105 Highland Pl., Ithaca, N. Y.
WEED, R. W. (Student, 1910), 404 N. Cayuga St., Ithaca, N. Y.

PENNSYLVANIA STATE COLLEGE

HASSLER, Joseph A. (Student, 1910), Alpha Kappa Delta House, Pa. State College, State College, Pa.
HOFFMAN, William S. (Student, 1910), 492 Main Bldg., Pa. State College. State College, Pa.

MATTERN, J. Fred (Student, 1910), Alpha Kappa Delta House, Pa. State College, State College, Pa.

MINSKER, John W. (Student, 1910), 339 McAllister Hall, Pa. State College, State College, Pa.

MORGAN, Henry (Student, 1910), 339 McAllister Hall, Pa. State College, State College, Pa.

PURDY, Donald F. (Student, 1910), 314 E. College Ave., Pa. State College, State College, Pa.

RAHN, Robert M. (Student, 1910), 370 Main Bldg., Pa. State College, State College, Pa.

WHITE, J. Frank (Student, 1910), 132 Beaver Ave., Pa. State College, State College, Pa.

UNIVERSITY OF ILLINOIS

BUTTERS, H. M. (Student, 1910), 210 E. Green St., Champaign, Ill.

BUYERS, D. E. (Student, 1910), 502 E. Green St., Champaign, Ill.

UNIVERSITY OF WISCONSIN

CHRISTIE, H. A. (Student, 1910), 229 W. Gilman St., Madison, Wis.

GRAY, C. F. (Student, 1910), 811 W. Johnson St., Madison, Wis.

SUHS, G. H. (Student, 1910), 225 W. Gilman St., Madison, Wis.

THOMPSON, O. T. (Student, 1910), 225 W. Gilman St., Madison, Wis.

COMING MEETINGS

APRIL–MAY

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AIR BRAKE ASSOCIATION

May 10–13, Dennisson Hotel, Indianapolis, Ind. Subjects for discussion, and chairmen: Air Brake Instruction, Examination and Rating, Thos. Clegg; Air Pump Piping, Fittings and Connections, George W. Kiehm; Best Arrangement of Air Pump and Main Reservoir Capacity for100-car Train Service, P. J. Langan; Brake Cylinders and Connections to Cylinder Leakage, W. P. Garabrant; Inspection and Cleaning of Triple Valves and Brake Cylinders, C. P. McGinnis; Developments in Air Brakes, W. V. Turner; New York Brake Equipment, T. F. Lyons; Westinghouse Equipment, S. G. Down; Recommended Practice, S. G. Down. Secy., F. M. Nellis, 53 State St., Boston, Mass.

AMERICAN ASSOCIATION ELECTRIC MOTOR MANUFACTURERS

May 18, Newport News, Va. Secy., Frank H. Couch, Hampton.

AMERICAN ASSOCIATION OF LOCAL FREIGHT AGENTS

April 19–22, Mobile, Ala. Secy., G. W. Dennison, Toledo, O.

AMERICAN ELECTROCHEMICAL SOCIETY

May 5–7, Spring Meeting, Pittsburg, Pa. Addresses on the Present Status of Electrochemical Industries; Pittsburg as an Electrochemical Center; the Conservation of Natural Sources of Power. Secy., Dr. J. W. Richards, Lehigh University, South Bethlehem.

AMERICAN EXPOSITION IN BERLIN

June 1–Aug. 31, American Manager, Max Vieweger, 50 Church St., New York.

AMERICAN FOUNDRYMEN'S ASSOCIATION and AMERICAN BRASS-FOUNDERS' ASSOCIATION

May 18–20, joint convention, Cincinnati, O. Secy., A. F. A., Richard Moldenke; A. B. F. A., W. M. Corse.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

March 30–April 1, Charlotte, N. C. Papers: Electric Drive in Textile Mills, A. Milmow; Gas Engines in City Railway and Light Service, E. D. Latta, Jr.; Modifications of Hering's Laws of Furnace Electrodes, A. E. Kennelly; The Proportioning of Electrodes for Furnace Electrodes, Carl Hering; Some Demonstrations of Lightning Phenomena, E. E. F. Creighton; Economics of Hydroelectric Plants, W. S. Lee; A Method of Protecting Insulators on the Lines of the Niagara and Lockport Power Company, L.

C. Nicholson. April 21, San Francisco, Cal. Papers: Economics of a Generator Power System, P. M. Downing; Hydroelectric Developments and Irrigation, J. C. Hays. July 31, annual meeting., 29 W. 39th St., New York. Secy., R. W. Pope, 29 W. 39th St.

AMERICAN MATHEMATICAL SOCIETY

April 30, Columbia University, 150 W. 116th St., New York. Secy., F. N. Cole.

AMERICAN PORTLAND CEMENT MANUFACTURERS

April 12. Secy., Percy H. Wilson, Philadelphia, Pa.

AMERICAN RAILWAY ASSOCIATION

May 18, New York. Secy., W. F. Allen, 24 Park Pl.

AMERICAN RAILWAY INDUSTRIAL ASSOCIATION

May 10, Memphis, Tenn. Secy., Guy L. Stewart, S. W. Ry., St. Louis, Mo.

AMERICAN SOCIETY OF CIVIL ENGINEERS

April 6, 20, 220 W. 57th St., New York. Papers, April 6: New York Tunnel Extension of the Pa. R. R.; The Terminal Station, West, B. F. Cresson, Jr.; The Bergen Hill Tunnels, F. Lavis. Papers, April 20: Federal Investigation of Mine Accidents, Structural Materials and Fuels at Pittsburg Testing Station, H. M. Wilson.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

April 9, St. Louis, Mo., with St. Louis Section, A. I. E. E., and Engineers Club of St. Louis. April 12, 29 West 39th. St., New York. April 27, Auditorium Edison Electric Illuminating Co. of Boston, Boston, Mass., Boston Section, A. I. E. E., and Boston Soc. C. E., coöperating. May 31–June 3, Spring Meeting, Atlantic City, N. J. July 26–29, meeting in Birmingham and London, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SUPPLY AND MCHY. MFRS. ASSOC. and SOUTHERN SUPPLY AND MCHY. DEALERS ASSO.

April 5–7, Convention, Seminole Hotel, Jacksonville, Fla.

AMERICAN WATER WORKS ASSOCIATION

April 26–30, annual convention, New Orleans, La. Paper: New Orleans Sewerage and Water Supply Systems, G. C. Earl. Secy., J. M. Diven, 14 George St., Charleston, S. C.

BROOKLYN POLYTECHNIC STUDENT SECTION, AM. SOC. M. E.

April 9. Paper: Engineering and Common Sense, William Kent, Mem. Am.-Soc. M.E. Secy., Percy Gianella.

CANADIAN FREIGHT ASSOCIATION

April 14, annual meeting, Montreal. Secy., T. Marshall, Toronto, Ont.

FLORIDA ELECTRIC LIGHT AND POWER ASSOCIATION

April 12, annual meeting, Tampa. Secy., G. I. Doig, Gainesville.

INTERNATIONAL MASTER BOILERMAKERS' ASSOCIATION

May 24–27, New Clifton Hotel, Niagara Falls, Ont. Secy., Harry D. Vought, 95 Liberty St., New York.

INTERNATIONAL RAILWAY FUEL ASSOCIATION

May 23–26, Chicago. Secy., D. B. Sebastian, 327 LaSalle St.

IOWA ELECTRICAL ASSOCIATION SHOW

April 20–21, Sioux City. Secy., W. N. Keiser, Des Moines Electrical Co., Des Moines.

IOWA STREET AND INTERURBAN ASSOCIATION
April 20, 21, Sioux City. Secy., L. D. Mathes, Dubuque.

MISSOURI ELECTRIC AND GAS ASSOCIATION
April 14–16, Jefferson City. Secy., C. L. Clary, Sikeston.

MODERN SCIENCE CLUB
April 12. Annual election, 125 S. Elliott Pl., Brooklyn, N. Y. Secy., J. A. Donnelly.

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS
April 27, 28, annual meeting, Boston. Secy., Dr. C. J. H. Woodbury, Mem.Am.Soc.M.E., Box 3772.

NATIONAL ASSOCIATION OF MANUFACTURERS
May 16–18, New York. Secy., George S. Boudinot, 170 Broadway.

NATIONAL DISTRICT HEATING ASSOCIATION
May, annual meeting, Toledo, Ohio. Secy., A. C. Rogers.

NATIONAL ELECTRIC LIGHT ASSOCIATION
May 23–28, St. Louis, Mo. Secy., Frank H. Tate, Dayton, O.

NATIONAL GAS ASSOCIATION OF AMERICA
May 17–19, Oklahoma City, Okla. Secy., M. W. Walsh, 110 N. Broadway.

NATIONAL MACHINE TOOL-BUILDERS ASSOCIATION
May 24, 25, Spring Convention, Hotel Seneca, Rochester, N. Y. Secy., C. E. Hildreth, Worcester, Mass.

NATIONAL METAL TRADES ASSOCIATION
April 13, 14, annual convention, Hotel Astor, New York.

NEW ENGLAND WATERWORKS ASSOCIATION
April 13, special meeting, Hartford, Conn. June, Providence, R. I. September 14–16, annual convention, Rochester, N. Y. Secy., Willard Kent, Narrangansett Pier, R. I.

OHIO SOCIETY OF ENGINEERS
May 19, 20, Cincinnati. Secy., F. E. Sanborn, Ohio State University, Columbus.

PENNSYLVANIA STATE GAS ASSOCIATION
April, Easton. Secy., W. H. Merritt, Lebanon.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
April 26, West Hall, R. I. School of Design, 8 p.m. Paper: Oxy-Acetylene Welding and Cutting, Henry Cave; May 24, Modern Machine Tools. Secy., Prof. T. M. Phetteplace, Mem.Am.Soc.M.E., 48 Snow St.

SOCIETY OF CHEMICAL INDUSTRY
April 1, annual meeting, New England Section. Secy., Alan Claflin, 88 Broad St., Boston, Mass.

STEVENS ENGINEERING SOCIETY
April 5, 12, 19, 26, Hoboken, N. J. Papers: Theory of Gyroscopic Motion, L. A. Martin, Jr.; Handling Concrete Work, F. B. Gilbreth, Mem.Am.Soc.-M. E.; Notable Examples in Modern Construction, J. C. Ostrup; Development of the New Navy, D. W. Taylor.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
April			
2	Amer. Soc. Hungarian Engineers and Architects	Z. deNemeth	8.30
7	Blue Room Engineering Society	W. D. Sprague	8.00
12	The American Society of Mechanical Engineers	C. W. Rice	8.15
12	Amer. Soc. Engineering Contractors	D. J. Hauer	2.30 8.00
14	Illuminating Engineering Society	P. S. Millar	8.00
15	New York Railroad Club	H. D. Vought	8.15
15	American Institute of Electrical Engineers	R. W. Pope	8.00
19	New York Telephone Society	T. H. Lawrence	8.00
27	Municipal Engineers of City of New York	C. D. Pollock	8.15
29	American Institute of Electrical Engineers	R. W. Pope	8.00

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1910

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John E. Sweet
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On Constitution and By-Laws

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Univ. of Maine, Orono, Me.	February 8	....	....	....

FREIGHT TRAIN RESISTANCE

ITS RELATION TO AVERAGE CAR WEIGHT

By Prof. Edward C. Schmidt, Urbana, Ill.

Member of the Society

CONTENTS

Train resistance varies not only with the train speed, but also with the average weight of the cars of which the train is composed. At a given speed the tractive effort required for each ton of weight of the train will be greater, for example, for the train which is composed of cars of twenty tons average gross weight, than for the train composed of cars which weigh, on the average, fifty tons each.

2 While this fact has been known for some years, it has found inadequate expression and but little application. In the establishment of their tonnage ratings many railroads have altogether ignored it. In the tonnage ratings of a few roads this variation of resistance with car weight is recognized to the extent of allowing a difference in rating between trains composed of loaded cars and those consisting entirely or partially of empty cars. Generally in such systems a certain amount is allowed arbitrarily, to be added to the weight of empty cars in determining, for the purpose of rating, the weight of the train in which they are found. In such rating no distinction is made between loaded cars of various weights, although such weights vary from 25 to 70 tons.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

3 A still smaller group of railroads have fully recognized the significance of the facts above stated in establishing their tonnage ratings, which in such cases are usually termed "adjusted" or "equated" ratings. Under these adjusted ratings the actual weight of the train allotted to a particular locomotive varies according to the number of cars in the train. The rating for the same locomotive, with trains of 40, 60, and 80 cars, for example, will be different in each of the three cases. This, of course, is in effect a variation of the rating with respect to the average car weights.

4 Most of these adjusted ratings have been empirically determined. In the few cases where they rest upon experiments made to determine the variations in train resistance with respect to car weight the data and results of such experiments have not been fully published. Existing train resistance formulæ likewise fail in most cases to take into account these variations of resistance with car weight; and probably much of the divergence among them is properly to be ascribed to this fact.

5 *Purpose of the Tests.* In view of the facts just stated it has seemed desirable to make the tests whose results are here recorded. They were planned to determine the resistance of freight trains under the usual conditions of operation; and were designed to disclose at the same time, if possible, the relation existing, at any given speed, between train resistance and average car weight. Since the chief use of such information is in the production of locomotive ratings, the conditions of the tests have been made like those which prevail in normal freight train operation. The speed range, for example, is from 5 to 35 miles per hour; and the trains experimented upon were trains in regular service, and usual in their make-up. The track upon which the tests were made is believed to be representative of good main line construction.

6 The tests have been made as part of the research work of the Engineering Experiment Station of the University of Illinois, conducted by the railway engineering department. They were begun in April, 1908, and were completed in May, 1909 All tests were made by means of Test Car 17, a dynamometer car, owned jointly by the University of Illinois and the Illinois Central Railroad, and were carried out on the Chicago division of this road.

7 In the preparation of the report the aim has been to present in it as brief a statement of the results and conditions as is compatible with a clear understanding of the tests. The results of the tests will also be published as a bulletin of the Engineering Experiment Station

of the University of Illinois. This bulletin will contain, in addition to the facts here presented, more detailed information concerning the track, the dynamometer car, and the methods of calculation; as well as the tonnage record for each train and the results and resistance curve for each test. A summary of the test conditions and the conclusions is inserted immediately beyond.

8 Throughout the report the terms "resistance" and "train resistance" mean the number of pounds of tractive effort required for each ton of the train, in order to keep it in motion on straight and level track, at uniform speed, and in still air. The report deals exclusively with the resistance of the train behind the locomotive tender. Locomotive and tender resistance are not discussed.

9 *Acknowledgments.* The tests have been made possible through the interest and coöperation of Mr. William Renshaw, Mr. J. G. Neuffer, and Mr. R. W. Bell, who were successively superintendents of machinery of the Illinois Central Railroad during the period of planning and conducting the work. Many other officials of the Chicago division of the road have rendered generous assistance in the investigation, which has entailed for them not a little inconvenience and labor. Such interest and assistance are thoroughly appreciated by those of the university staff who have been concerned with the work.

10 Throughout this work, the operation of the dynamometer car and the making of the calculations have been under the direct supervision of Mr. F. W. Marquis, Associate in the railway engineering department, Engineering Experiment Station. Much of whatever accuracy and reliability have been attained in the investigation is due to his intelligent and painstaking care in making the tests and in systematizing the work of calculation. He has also rendered great assistance in supervising the preparation of the tables and illustrations, and in the final checking of the manuscript.

SUMMARY AND CONCLUSIONS

11 *Summary.* The report deals with the results obtained from tests of thirty-two ordinary freight trains, whose chief characteristics were as follows:

	Minimum	Maximum
Total weight, tons	747	2908
Average weight per car, tons	16.12	69.92
Number of cars in the train	26	89

The trains whose average weights were less than 20 tons or more

than 60 tons were composed of cars of nearly uniform weight; while those whose average car weights were between 20 and 60 tons were either homogeneous or mixed as regards the weight of the individual cars.

12 The tests were made during generally fair weather. The minimum air temperature during any test was 34 deg., the maximum 82 deg. The approximate average wind velocity prevailing throughout one test was 25 miles per hour; during all the others it was less than 20 miles per hour.

13 The tests were made upon well constructed and well maintained main-line track, 94 per cent of which is laid with 85-lb. rail, the remainder being laid with 75-lb. rail. Except through station grounds, where screenings or cinders are used for ballast, the track is full ballasted with broken stone.

14 *Conclusions.* The results of the tests are presented in Figs. 10 and 11, in Table 3, and in the equations accompanying Par. 75. The curves, the table, and the equations, are each different expressions of the same facts. It is believed that by their use one may safely predict the probable total resistance of *entire* freight trains at various speeds, when running upon straight and level track of good construction, during weather when the temperature is above 30 deg. fahr., and the wind velocity not more than 20 miles per hour: provided the *average* weight of the cars composing the train be known.

15 The results are applicable to trains of all varieties of make-up to be met with in service. They may be applied, without incurring material error, to trains which are homogeneous and to those which are mixed, as regards individual car weight.

16 The results are primarily applicable to trains which have been for some time in motion. When trains are first started from yards, or after stops on the road of more than about twenty minutes duration, their resistance is likely to be appreciably greater than is indicated by the results here presented. In rating locomotives no consideration need be given this matter, except in determining "dead" ratings for low speeds, and then only when the ruling grade is located within six or seven miles of the starting point or of a regular road stop.

17 It is to be expected that some trains in service will have a resistance about 9 per cent in excess of that indicated by Figs. 10 and 11, due to variations in make-up or external conditions within the limits to which the tests apply. If operating conditions make it essential to reduce to a minimum the risk of failure to haul the allotted tonnage, then this 9 per cent allowance should be made. This con-

sideration, like the one preceding, is important only in rating locomotives for speeds under 15 miles per hour. At higher speeds, the occasional excess in the resistance of individual trains will result in nothing more serious than a slight increase in running time. It should be emphasized that this allowance, if made, is to be added to the resistance on level track—not to the gross resistance on grades.

THE METHODS AND MEANS EMPLOYED IN CONDUCTING THE TESTS

18 The tests were carried on by means of the dynamometer car here referred to as Test Car 17, which, when not in use, is held at Champaign, a district terminus. The car was operated from time to time in the regular trains leaving this point, and the trains selected were partly in the northbound, partly in the southbound traffic.

19 The plan was to determine, for each of the trains experimented upon, the relation of its resistance to its speed. This information was to be finally expressed as a resistance-speed curve such as is shown in Fig. 1. The trains were so selected that their average car weights varied through as great a range as possible. As will later appear, this range proved to be from the weight of an empty gondola to that of a fully loaded car of 100,000-lb. capacity. It was the expectation that when the resistance-speed curves of the individual tests were brought together, their analysis would reveal the relations existing between train resistance and car weight.

20 During each test the following information was obtained:

a The drawbar pull of the locomotive upon the train.
b The train speed.
c A continuous record of time elapsed from the beginning of the test.
d The pressure existing in the brake cylinder of the test car.
e The direction of the wind relative to the direction of motion of the car.
f The velocity of the wind relative to the car.
g A record of the location of the test car upon the road.
h Air temperatures and other weather conditions.
i Data concerning the train, such as its weight, etc.

The information cited under Items *a* to *g* was obtained in the form of continuous graphical records upon the chart which is produced by the apparatus of the dynamometer car. By means of this chart any of the quantities mentioned may be determined at any point upon the road.

21 The curves of drawbar pull and speed provide the informa-

tion essential to the investigation. Supplemented by an accurate profile and a record of train weight, they enable net train resistance to be calculated at any position of the train upon the road. The time record provides a means of calibrating and checking the speed curve. The pressure in the brake cylinder was recorded merely to make it possible to distinguish those periods during the test when the brakes were applied to the train; it being obviously necessary to ignore such portions of the record when making the calculations.

22 The relative wind velocity and relative wind direction were obtained by means of an anemometer and a wind vane mounted on the roof of the test car. When compounded with the known speed and direction of motion of the car, these data permit the determination of the actual wind direction and wind velocity with respect to the track, which were recorded in each test for each point at which train resistance was determined. It is probable that these wind data are, under some circumstances, subject to a considerable error. Considering the length of the run made with each train and the length of time it was on the road, it is believed the data are, nevertheless, more reliable than those which might have been recorded by stationary instruments located at one or two points along the track.

23 Item *g*—the location of the car upon the road—was defined by marking upon the test-car record the position of mile posts and stations at the moment they passed the car. By means of this record it is possible to correlate any position of the train with the road profile.

24 Data concerning the train were obtained by one or two observers who had no other duties. With the one exception noted beyond, all trains were weighed to determine their tonnage. In addition to its weight, there was recorded for each train, its length[1] and for each car its number, kind, stenciled "light weight," gross weight, capacity, and the initials of the owning road.

25 All test-car instruments were calibrated before the tests, and their calibrations were frequently checked during the progress of the investigation. All observers were men experienced in the operation of the test car, and many of them had participated also in the work of calculation, and were consequently aware of the points at which alertness and care were especially needed. No effort has been spared, in conducting the tests, to ensure accuracy in the data. These facts

[1] Train length was determined by counting, during the test, the number of rail lengths corresponding to the length of the train and multiplying this number by 30 ft., which is the rail length for this track.

are here mentioned as having some significance to one who may undertake to estimate the reliability of the results. The Appendix contains an illustration of one of the test-car charts and a description of the car itself.

26 This report includes the data and results from tests of thirty-two different trains. For the purposes of this research, tests were made of twelve other freight trains; but their results were finally excluded from the report. Three of these additional tests were rejected because of uncertainty about the train weights, one because of a breakdown in the test-car recording apparatus during the progress of the test, and eight were disregarded because the temperatures prevailing were below the range for which it was intended the results should apply; the low temperature in some cases being coupled with high wind.

TEST CONDITIONS AND TRAIN DATA

27 *The Trains Tested.* The test trains were all of such make-up as naturally resulted from the traffic conditions in the Champaign yards. For most of the tests the test car was simply coupled into the trains selected by the trainmaster, solely with reference to his convenience in operating and in returning the test car. As the investigation progressed, it became apparent that the accumulated data left certain gaps in the range of average car weights. There were at this stage, for example, few trains experimented upon with average car weights near 25 to 30 tons, and none with an average car weight of 70 tons. The last six or eight trains were therefore made up especially to supplement the data at these points. It should be understood, however, that nothing in this process resulted in a train-make-up which was in any respect unusual. All the trains tested are, therefore, such as one might expect to find upon any road where the traffic conditions are normal. They include trains made up almost entirely of empty gondolas,[1] others with considerable variation in both load per car and kind of car, and still others composed almost entirely of loaded box cars or of loaded gondolas.

28 Test S-1018 demands special mention in this connection. The train for this test included Illinois Central Railroad locomotives No. 423 and No. 732, weighing respectively 145,200 and 223,600 lb.

[1] Cars are designated as box, stock, gondola, flat and tank cars. The term box car is made to include refrigerator cars, the test car and the caboose. The term gondola includes all unroofed cars with sides, such as coal cars, hopper cars, etc.

Their combined weight constituted 13.6 per cent of the total train weight. These locomotives, with their tenders, were being hauled "dead" and had the main rods disconnected, as is usual in such cases. The first is of the 2–6–0 type, the second of the 2–8–0 type, and they and their tenders had therefore together seventeen axles in operation. For the purpose of determining the average car weight for this train, these two locomotives were assumed to be equivalent, in their resistance, to a number of cars having a like number of axles, i.e., $4\frac{1}{4}$ cars. The results of the calculations warrant the belief that this view of the situation has resulted in no material error. A study of Table 1 will make clear the diversity in the composition of the trains.

29 All trains, except Nos. S-1016, S-1018, S-1030A and S-1030B, were weighed upon one of the two track scales at Champaign. This weighing was done in the usual manner, by pulling the train over the scales and weighing the cars successively without uncoupling. These track scales were in good condition and were each inspected four times during the test period, these inspections disclosing a maximum error in one scale of $-\frac{1}{5}$ per cent, in the other of $-\frac{1}{2}$ per cent. The train in Test S-1016, composed entirely of empty cars, by an error in arrangement left the yards without being weighed. The weights stencilled on the cars were accepted as correct in this case. The train in Test S-1018 was weighed upon track scales in the Chicago yards; and the trains of Tests S-1030A and S-1030B were weighed in the yards at Centralia. In Test S-1021, after leaving the yards, two cars were added to the train, for which the weights were determined from the stencilled weights and the way-bills. In Tests S-1030B and S-1048 the weights of one and two cars respectively were similarly determined, and in Test S-1061 the stencilled weight was used for one empty car. Obviously no important errors in the total tonnage have resulted from possible inaccuracies in the weights of these cars.

30 All cars of all trains were of course provided with the usual four-wheeled truck. Presumably the majority of the cars had journals conforming to the specifications of the Master Car Builders' Association, which for some years have required that freight car journals be either $3\frac{3}{4}$ in. by 7 in., $4\frac{1}{4}$ in. by 8 in., 5 in. by 9 in., or $5\frac{1}{2}$ in. by 10 in. in size, depending upon the car capacity. It is safe to assume that all trucks were provided with wheels of 33 in. standard diameter.

31 Throughout each test, observations were repeatedly made to discover such irregularities as hot journal boxes, brakes which were not free from the wheels, and trucks which did not freely follow the track. Such things occurred to the usual extent; a hot-box or two or

an unreleased brake being occasionally found on some of the trains, while others were entirely free from such defects. The record of such matters was given consideration in making the calculations; but, as was anticipated, the results showed no discrepancies which could be explained by such causes.

32 The range over which the train data for all of the tests varied is as follows:

	Minimum	Maximum
Total train weight, tons	747	2908
Average weight of cars composing the train, tons	16.12	69.92
Number of cars in the train	26	89
Train length, ft	1120	3480

33 *The Track.* The track upon which the experiments were carried on extends from Gilman to Mattoon, Ill., a distance of 91 miles, and lies upon the Chicago division of the main line of the Illinois Central Railroad. Until about 10 years ago this was a single-track road and one of the oldest in the State. At that time a second track was constructed, and the roadbed for both tracks is now well settled and in good condition. The maximum grade against northbound traffic is 29 ft. per mile, and against southbound traffic, 31.9 ft. per mile. In all the 91 miles there are only 7850 ft. of curved track.

34 Through station grounds the tracks are ballasted with screenings or cinders; all other portions of both tracks (about 83 of the 91 miles) are full ballasted with broken limestone. The cross-ties are of oak, laid 20 in. center to center. About $10\frac{1}{4}$ miles of the west track is laid with 75-lb. A.S.C.E. rail, put down in 1894 and 1895; while the remainder of the west track and all of the east track is laid with 85-lb. A.S.C.E. rail, the oldest of which was put down in 1900. During 8 months of the year there is employed in maintaining this portion of the road a force of men which averages one man per mile of track; during the other 4 months this force is reduced to one man for each 2 miles. As regards both construction and maintenance this track is such as one may expect to find upon the main lines of first-class railroads.

35 These 91 miles of track were especially surveyed, immediately preceding the tests, by the railway engineering department of the university for the purposes of this and similar investigations. The levels were run on the east track, and readings were taken to 0.1 ft. at stations 300 ft. apart; and turning points were taken at every fourth

TABLE 1 SUMMARY OF TEST CONDITIONS AND TRAIN DATA

TEST LAB. SERIAL NO.	TEST DATE	Weather Conditions						Train Data†										
			AIR TEMP. DEG. F.		AVE. APPROX WIND VELOCITY MILES PER HOUR	RANGE OF DIRECTION OF WIND WITH RESPECT TO TRACK*		TRAIN LENGTH FEET	WEIGHTS		CARS IN TRAIN TOTAL No.	Train Make-up						
												CONDITIONS OF LOADING			KIND OF CARS			
			Beginning of Test	End of Test		From	To		GROSS TRAIN WEIGHT TONS	AVERAGE GROSS WEIGHT PER CAR TONS		Empty Cars No.	Loaded Cars No.	Loaded Cars %	Box Cars %	Gondola Cars %	Flat Cars %	Tank Cars %
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
	1908																	
S-1013	4/27	Wet	42	44	19	+35°L	+90°L	2784	2549	38.04	67	10	57	85	82	13	0	5
S-1015	4/29	Fair	40	48	10	+25°L	−80°L	2520	2489	36.08	69	8	61	88	68	6	16	10
S-1016	4/30	Fair	44	48	10	+15°R	+60°R	3030	1161	16.12	72	72	0	0	3	97	0	0
S-1017	5/1	Wet	48	54	16	+45°L	−80°L	2670	2532	38.44	66	13	53	80	95	5	0	0
S-1018‡	5/2	Fair	40	45	11	+10°R	+85°R	2130	1353	25.40‡	49‡	34	15	31	61	6	16	17
S-1019	5/9	Fair	44	62	25	+20°R	+45°R	3480	1572	17.72	89	75	14	16	34	58	7	1
S-1021	5/13	Wet	66	70	17	+60°R	+80°R	2400	2908	46.16	63	10	53	84	32	60	5	3
S-1023	5/23	Fair	62	74	17	+75°R	−80°R	2320	2243	38.72	58	17	41	71	45	52	0	3
S-1027	7/2	Wet	64	80	14	−50°R	+70°R	1710	2185	47.44	46	3	43	94	22	76	2	0
S-1030A	7/8	Fair	60	68	6	+20°R	+65°R	1380	2036	59.88	34	2	32	94	6	94	0	0
S-1030B	7/8	Fair	68	72	7	+20°R	+65°R	1650	2342	57.12	41	3	38	93	20	80	0	0
S-1031	7/22	Fair	70	82	5	+ 0°	+40°L	1425	747	20.72	36	30	6	17	94	3	3	0
S-1033	9/26	Fair	66	82	12	+ 5°L	+15°R	1710	2275	51.70	44	2	42	95	5	95	0	0
S-1034	10/3	Fair	42	60	4	− 0°	+85°L	3015	1259	16.56	76	76	0	0	1	99	0	0
S-1036	10/10	Fair	40	62	6	+15°R	−15°R	2010	1961	37.72	52	8	44	85	73	25	2	0

S-1038	10/15	Fair	58	72	16	+ 5°L	+25°L	1650	2144	52.28	41	3	38	93	22	78	0	0
S-1040	10/24	Wet	57	53	11	+15°R	+40°R	1830	2152	45.76	47	2	45	96	49	49	0	2
S-1043	11/7	Fair	38	53	8	+ 5°R	+65°R	2580	1118	16.92	66	65	1	2	24	74	0	2
S-1048	11/28	Fair	36	39	6	+ 5°R	+30°L	A2175	2443	45.24	54	8	46	85	37	63	0	0
								B2100	2355	45.24	52	8	44	85	38	62	0	0
	1909																	
S-1050	1/23	Fair	53	66	8	+ 0°	−25°R	1620	1618	40.44	40	16	24	60	75	25	0	0
S-1052	1/28	Wet	36	40	11	−45°L	+70°L	2430	1514	24.80	61	44	17	28	61	38	1	0
S-1057	3/6	Fair	34	40	10	+20°R	−35°L	1830	2107	41.32	51	8	43	84	49	43	6	2
S-1061	3/13	Fair	41	38	7	+45°L	−85°L	1785	2252	51.20	44	3	41	93	5	84	11	0
S-1063	3/19	Wet	39	40	12	+20°R	+40°R	3060	1484	20.04	74	70	4	5	7	93	0	0
S-1070	4/17	Fair	58	71	4	+ 0°	−65°L	2400	1622	24.60	66	49	17	26	58§	42	0	0
S-1072	5/1	Fair	35	37	17	+70°L	+90°L	1200	1859	66.40	28	1	27	96	4	96	0	0
S-1073	5/4	Fair	53	63	10	+25°L	+70°R	1200	1880	67.16	28	1	27	96	4	96	0	0
S-1074	5/7	Fair	45	60	10	+65°L	−80°L	3180	1340	16.56	81	81	0	0	2	98	0	0
S-1076	5/11	Fair	51	67	16	+40°R	+75°R	1120	1818	69 92	26	1	25	96	4	96	0	0
S-1077	5/14	Fair	64	70	13	−25°R	−75°R	2145	1505	28.40	53	35	18	34	74	26	0	0
S-1079	5/18	Fair	65	68	18	+65°R	−85°R	2070	1685	33.04	51	14	37	73	90	10	0	0
S-1080	5/21	Fair	50	70	11	+ 0°	+45°L	2550	1347	21.40	63	57	6	10	16	84	0	0

* Direction is designated by the angle made with the track. A wind any component of whose velocity helps the train forward is marked +; winds with components of opposing velocity are marked −. Winds from the right side of the track are designated as R; from the left side as L. Thus +40°R means a wind blowing from the rear and from the right hand side, whose direction makes an angle of 40 deg. with the track.

† All data apply to the train only—engine and tender are excluded. In Columns 9 to 19, for test S—1048, A indicates, from Champaign to Rantoul, B from Rantoul to Gilman.

‡ Train in Test S-1018 had two "dead" locomotives and tenders in addition to cars noted.

§ This number included 15 stock cars—classed as box.

station where levels were read to 0.01 ft. The results of the survey are expressed in a profile drawn to a scale of $\frac{1}{4}$ in. to 100 ft., which was used in making the test calculations.

36 *The Weather Conditions.* In Table 1 the weather prevailing during each test is designated as either fair or wet, wet weather meaning either continuous or intermittent rain. During seven of the thirty-two tests the weather was wet. The lowest air temperature recorded at any time during any test is 34 deg. fahr. and the highest recorded temperature is 82 deg.

37 The column headed "average wind velocity," in Table 1, presents the averages of the calculated wind velocities derived for each point or section of the test in question for which the train resistance was determined. There was a considerable variation at different points during the same test. The approximate maximum average velocity prevailing during any test was 25 miles per hour, the minimum 4 miles per hour. The actual wind direction (with respect to the track) varied during the tests, as would be expected, through the entire 360 degrees.

38 It was intended so to select the tests that the weather conditions, the temperatures, and the wind velocities, would be such as usually prevail in most parts of the country from the middle of spring until the middle of autumn when the basic or "summer" tonnage ratings are in force—such conditions, in short, as would give rise to no appreciable difficulties in train operation.

METHODS EMPLOYED IN CALCULATING THE RESULTS

39 The immediate purpose in making the calculations was to produce for each test a curve showing the relation between resistance and speed, for as great a variety of speeds as the data would permit. This involves calculating the train resistance at various positions of the train upon the track, and the first step is the inspection of the test-car record in order to select suitable points or sections at which the resistance may be calculated. The considerations of first importance are, that the points should finally represent as great a speed range as possible, and that the speeds should be approximately evenly distributed within this range. Only points and sections where the entire train was running and continued to run upon straight track, were selected; resistance due to track curvature is therefore entirely eliminated. The data essential to the process of calculation are the draw-bar pull of the engine, the train speed and its acceleration, the ton-

nage, and the profile. The pull and the speed, as previously stated, were determined from continuous curves drawn on the test-car chart. Two processes were used, designated here as Method 1 and Method 2. By Method 1 the momentary values of pull, speed, acceleration, and grade, were determined for a particular position of the train upon the road; by Method 2 the average values of these quantities were determined for the period during which the test car was passing over a definite section of the track.

40 *Method 1, Resistance at a Point on the Road.* The point having been chosen, the pull and the speed were found by direct readings from the chart. This pull divided by the tonnage gives the gross train resistance at this speed. This gross resistance was next corrected for both acceleration and grade resistance. The acceleration was determined, by graphical methods from the speed curve, and the grade was found by correlating the train's position with the profile. The points were all so selected that at the moment under consideration, the entire train was on a nearly uniform grade. Method 1 results in momentary values of train resistance at the points considered.

41 *Method 2, Average Resistance over a Section.* By this method the average value of train resistance was determined for the period during which the test car at the head of the train was passing a selected section of the track. This track section, corresponding to a certain length or section on the test-car record, was so selected that the speed of the car when entering was nearly equal to its speed when leaving it; and further so that no considerable variations in speed occurred during transit over the section. The sections chosen have varied in length from about ¼ mile to 1 mile. The variations in speed in passing the section have generally amounted to less than 2.0 miles per hour, and the maximum variation over any selected section is 11.7 miles per hour. In only 58 cases out of a total of 560 does this speed variation exceed 5.0 miles per hour.

42 These portions of the chart being chosen, the average pull was next found by determining the average ordinate of the curve of drawbar pull, and the average speed was found by means of the section length and the time record. Gross resistance in pounds per ton was next derived by dividing this value of pull by the tonnage, and this gross resistance was then corrected for the resistances due to acceleration and grade as in Method 1.

43 In this case the average acceleration was found by consideration of the speeds at entrance to and exit from the section. In order

to correct for grade, the elevation of the center of gravity[1] of the train was determined for that position of the train at which the test car entered the section, and again for the position at which the car left the section. The difference between these elevations establishes the effective average grade, which either helps or opposes the locomotive while the train passes the section. These elevations of the center of gravity of the train may not be determined with sufficient accuracy unless the train at the moment, is on a practically uniform grade. The section limits were therefore so chosen.

44 Method 2 results in a value of *average* train resistance for the *average* speed at which the train passes the section under consideration. It would be rigidly correct if train resistance varied uniformly with speed, in other words, if the curve showing the relation of resistance to speed were a straight line. This, of course, is not the case, and the process therefore gives results which are slightly in error. However, as stated above, the section was so chosen that the difference between the speeds at entrance to and exit from the section was small; and for the speed range represented by this difference, the curve of train resistance deviates but little from a straight line. Such error as does result from the process is, therefore, very small and of no moment whatever when compared with the variations due to natural causes, that occur in the resistance itself.

45 *Comparison of the Two Methods.* The two methods are fundamentally alike. Although the first is the less laborious, it requires the determination of acceleration at a point on the speed curve; which it is sometimes difficult to make accurately. For this reason the second method is generally preferable. Method 2 also deals with average values and therefore tends to eliminate from the results the incidental momentary variations in resistance. Consequently the second method has been employed whenever possible, and the first method generally resorted to only in cases where the limitations imposed in the selection of sections for Method 2 would have resulted in too few values from which to plot the resistance curves. Of all the individual resistance values incorporated in the report only 32 per cent were determined by Method 1. The care exercised in the calculations, and a study of the plotted values obtained by both processes, seem to

[1] The location in the train of its center of gravity was determined thus: Assume a train which weighs 1800 tons, is 2400 ft. long, and is composed of 60 cars. By inspection of the tonnage record we find that one-half of this weight (900 tons) lies in the first 25 cars. Hence the center of gravity is located $\frac{25}{60} \times 2400 = 1000$ ft. from the front end.

warrant the conclusion that their results are equally reliable. In Fig. 1 the circles represent values derived by Method 1, and the circular black spots represent values obtained by Method 2.

46 *General Considerations.* Even in freight train operation the tractive effort required to produce acceleration in speed is frequently greater than that required to overcome all other resistances combined. To produce, for example, an acceleration of 0.1 miles per hour per sec., requires a tractive effort of about 9 lb. per ton, in addition to that required by net train resistance and grade resistance. Since the acceleration resistance may constitute so large a proportion of the gross resistance, it is important that its determination be made with great care. This fact has been impressed upon all concerned with these tests. In calculating the acceleration resistance, both the force required to produce acceleration in the rotation of the wheels and axles, and the force required to produce the acceleration in the motion of translation of the train as a whole, were determined.

47 The test-car records make it possible to distinguish those portions of each test where the brakes were applied. Such places, few in number, were of course avoided in selecting points and sections for determining resistance. The records also show where hot-boxes and unreleased brakes were discovered in the train, and such defects were given consideration in making the calculations. They occurred infrequently and their effect could not be distinguished in the results. While therefore such portions of the record were avoided if convenient, sections and points on the charts otherwise suitable for calculation, were not rejected on these accounts.

48 *Effect of Stops in Limiting the Selection of Points and Sections.* Early in the progress of this work, when low air temperatures were first encountered, it became apparent that when the train was first started from rest, its resistance, calculated for a number of points at which the speed was the same, occasionally was unusually high. This was true not only for those portions of the run made immediately after leaving the yards; but also for those portions immediately following stops on the road. In a certain test, for example, the values of net resistance, calculated at various points at all of which the speed was 20 miles per hour, varied between 6.8 lb. and 5 lb. per ton—a difference of 27 per cent—for points selected within the first 9 miles of the run; whereas values of resistance at the same speed, determined later in the test, differed by only 10 per cent. The air temperature during this test (not included in the report) varied between 22 deg. and 26 deg.

49 For a number of tests such resistance values were plotted with respect to the distances from the yards of the points to which they apply. This process disclosed a surprisingly regular decrease in the resistance, until a distance of approximately ten miles was reached, after which the resistance had settled down to a fairly uniform value. Similar variations were found to occur to some extent during tests when the air temperature was as high as fifty or sixty degrees. This study[1] led to the conclusion that the difference in resistance was due to variations in the conditions of lubrication of the car journals, and that such variations were chiefly caused by changes in journal temperature. All this is, of course, in accord with the common belief of those experienced in train operation. The reason for discussing it in this place is that the facts stated have influenced the procedure in making calculations for this series of tests.

50 Since the variations in resistance are so great during the early part of the run no point or section within about the first ten miles has been selected for calculation in any test. If other points or sections, located farther from the start, were near stops, such points were rejected unless further investigation proved that at these places the train resistance had become nearly uniform in value. Fortunately operating conditions were such as to entail few stops, and the selection of points and sections for the calculations has not been unduly limited on these accounts.[2]

51 The effect of these limitations is to make the results of this investigation primarily applicable to trains which have been for some time in motion. Since, however, stops are not usually made upon ruling grades; and since if stops are made at other places on the road, the locomotive has available tractive power in excess of the requirements, the results of these tests are generally applicable in the solution of tonnage rating problems, except where the ruling grade occurs near a yard or other point where the trains are made up. In such cases the tonnage determined from the resistance curves here presented may prove to be somewhat too great.

52 *The Derivation of the Resistance Curves.* The calculations

[1] Further investigation of this matter is in progress, and the results are likely soon to be published.

[2] During the thirty-two tests included in the investigation, only sixty-eight stops, all told, were made after leaving the yards. Of these, one was of 55 minutes duration, nine lasted between 20 and 40 minutes, twenty-two between 10 and 20 minutes, and thirty-six less than 10 minutes.

result, for each test, in a series of values of net train resistance at a variety of speeds. These values of resistance were plotted with respect to speed, and gave such a diagram as is shown in Fig. 1. The curve, such as is there shown, was next drawn to express, for the test in question, the relation existing between resistance and speed. In order to draw this curve, the plotted points were assumed to be arranged in a number of groups, and for each group the averages of the values of speed and of resistance were determined. By these averages a new point or "center of gravity" of the group was then plotted. The curve was drawn by confining attention to the few points thus determined. The groups of points were arbitrarily selected so that the resulting "centers of gravity" would be nearly equidistantly distributed throughout the speed range. All curves presented in the report, except those exhibited in Fig. 11, were drawn by this process.

53 All reasonable precautions have been taken to attain accuracy in the calculations. In determining each value of resistance, each step was duplicated at a different time, and generally by a different person. The transcription of all tables, the plotting of points and the drawing of curves, have been similary checked.

RESULTS OF THE TESTS

54 *Results of the Individual Tests.* The immediate result of each test is a curve which shows for the train under consideration, the relation existing between train resistance and speed. Fig. 1 is such a curve derived from test S-1021. It is fairly representative of the entire group of curves, and such discussion of it as follows is general in its application.

55 The plotted points show unmistakably an increase in resistance as the speed increases, and the curve drawn represents the mean relation between resistance and speed. In Fig. 1 the maximum variation from this mean of any calculated value of resistance is about 20 per cent; the next largest variation is 16 per cent, and other calculated values of resistance generally differ from the values determined from the curve by less than 10 per cent. In a majority of the tests the maximum variation is less than in Fig. 1, and the general agreement between the calculated values of resistance and the ordinates of the curve is better than in the test chosen for illustration.

56 It has been thought desirable to express more specifically this variation between the calculated values of resistance and the mean values as derived from the curves drawn. To this end, for all tests,

all calculated values of resistance for speeds between 8 and 12 miles per hour were compared with the ordinates of the curves at the corresponding speeds, and the percentage difference was determined in each case. These percentages were then arranged in two groups and averaged. The one group included the results from all points lying above the curve, the other from those lying below it. The whole process was next repeated for speeds between 28 and 32 miles per hour, with results as follows:

Average Deviation (for all tests) of Calculated Resistance from the Mean Values Derived from the Curves, Expressed in Percentage of the Mean Values.

Speed	Above the Mean	Below the Mean
8 to 12 m.p.h.	6.4 per cent	7.6 per cent
28 to 32 m.p.h.	5.6 per cent	6.6 per cent

Such variation seems not extraordinary for this class of experimental work.

57 These differences may be due in part to accumulated errors in the instruments or in the calculations. In all cases, however, where the calculated value of resistance varied by an unusual amount from the mean, all calculations leading thereto were repeated and errors thus discovered are eliminated from the report. The explanation for such differences need not be sought further than in the variations which actually occur from time to time, in the resistance itself. Variations in such components of train resistance as flange friction and wind resistance are probably sufficiently great to account for the differences discussed above. The data do not permit the influences of such components of resistance to be differentiated.

58 The curve drawn for each test has been accepted as representing the average values of net train resistance, with a degree of accuracy sufficient for the purpose of rating locomotives. Such temporary excess of resistance as may be expected to occur will generally be absorbed in that reserve in the tractive effort of the locomotive which must be allowed in any system of tonnage rating.

59 *Results of all the Tests.* The resistance curves for the individual tests have all been brought together on one sheet, a reproduction of which is shown as Fig. 2.* Fig. 2 displays the immediate results

* The numbers shown on the curves are the last two figures of the test numbers. The curve marked 43 is derived from Test S-1043.

of the whole research. The lower curves give values of resistance varying from 3 to 5½ lb. per ton, while the upper curves show resistance values varying from 7 to 14 lb. per ton. Resistance values at the lower speeds differ by 100 per cent, and values at higher speeds differ by as much as 200 per cent.

60 If further analysis had not revealed the cause of the great variation in resistance here shown, Fig. 2* would have remained a useless exhibit. The explanation of this variation was sought in the test conditions enumerated below:

a Weather and temperature conditions.
b Wind velocity and direction.
c Kind of cars composing the train.
d Position of the loaded cars in the train.
e Defects in train equipment.
f Average weight of the cars in the train.

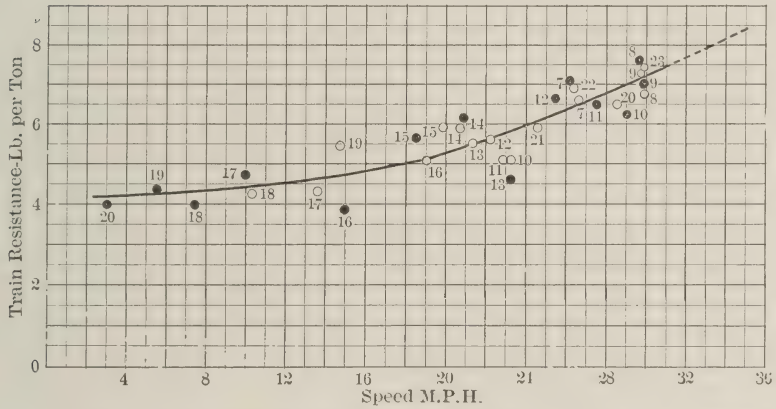

Fig. 1. Relation of Resistance to Speed for Test S-1021

AVERAGE WEIGHT PER CAR 46.16 TONS

61 The first five conditions either are uncontrollable or purposely were not controlled during these experiments, and attempts to explain the differences by reference to one or the other of them have been altogether unsuccessful. While difference in wind velocity, for example, might be a plausible explanation of the differences between two or

*Table 2 has been prepared from the original curves of the individual tests, only one of which is separately presented in the report (see Fig. 1). It gives no information not obtainable from Fig. 2; but presents the information in more convenient form, since the number of curves drawn in the figure makes it confusing.

three curves selected at random from Fig. 2, such explanation will not hold when applied to two or three other curves similarly chosen; and it fails altogether to explain such differences when it is applied to the whole group. The same remarks apply to any other of the first five items.

62 Item *f*, however, has furnished the clue whereby the apparent confusion in the results of the tests, as exhibited in Fig. 2, has been explained. It may be stated at once that the difference in train resistance for various tests is believed to be chiefly due to differences in the average gross car weights existing during the tests. An explanation of the process which led to this opinion follows immediately below. As was stated at the outset, this conclusion was anticipated when the work was begun, and the average car weight was therefore controlled during the experiments, and made to vary through the widest possible range.

63 *The Effects of Car Weight on Resistance.* The four upper curves of Fig. 2 are derived from trains in which the average weight per car was about 16 or 17 tons. The lowest curves are those derived from trains in which the car weight was nearly 70 tons. These facts serve as a rough indication of the part played by car weight in effecting changes in train resistance. This influence is more definitely brought out in the following discussion.

64 If from each of the curves of Fig. 2 the value of resistance is determined at one speed, say 5 miles per hour, these values of resistance may then be plotted with respect to their corresponding values of car weight; and since the speed is common, its influence is eliminated and the resulting diagram may be expected to reveal the relation existing between train resistance and average weight per car. Table 2 was prepared to facilitate this process. In it the tests are arranged in the order of the average car weights, which are given in the second column, and in the succeeding columns are set down the resistance values obtained from the *curves* of the individual tests, for each of seven different speeds. Table 2 therefore presents the values of the coördinates of seven points on each of the curves of Fig. 2, and hence, like Fig. 2, summarizes the immediate results of all tests.

65 In Table 2 the second and third columns present a series of values of average car weight and of train resistance at 5 miles per hour. Each pair of values represents the results of one of the 32 tests. Using these pairs of values as coördinates, a series of points has been plotted to form a new diagram, Fig. 3. For example, the point marked 21 in Fig. 3 is derived from the curve of Test S-1021.

The curve of resistance for this test (see Fig. 1 or Fig. 2) shows that at 5 miles per hour the mean resistance is 4.21 lb. per ton. During this test the average weight of the cars in the train was 46.16 tons. Table 2 also exhibits both of these values which, when plotted in Fig. 3, determine the point there marked 21. The other points of Fig. 3 were similarly determined. Each point represents the value of resistance at 5 miles per hour, derived from a particular test train.

66 Although there is considerable variation among the points of Fig. 3, they indicate clearly a decrease in the resistance as the car weight increases. The curve drawn in Fig. 3 represents, for the trains tested, the mean relation which existed, between resistance at 5 miles per hour and the average car weight.[1] For higher speeds this relation between resistance and car weight is shown by Figs. 4 to 9, which were derived by the same methods employed in producing Fig. 3.

67 The variation in resistance represented by the points in Figs. 3 to 9 is sufficient to warrant further discussion. Such discussion will, however, be postponed until later in the report. The conclusion reached is that these variations are largely caused by factors which are uncontrollable in ordinary train operation. If this be admitted, it is clear that the discussion of such variations may enter into the solution of tonnage rating problems, only as an argument for reserve tractive effort in the locomotive. An estimate of the desirable amount of such reserve appears beyond.

68 The curves of Figs. 3 to 9 have been accepted as representing, for these tests, the mean relation which existed betwe entrain resistance and the average gross weight of the cars composing the trains. These curves exhibit this relation at seven different speeds, 5, 10, 15, 20, 25, 30, and 35 miles per hour. For convenience in use, and to make comparison easier, these seven curves have been brought together in one diagram which is reproduced in Fig. 10.

69 Fig. 10 presents the final results of the whole research. Each of the curves there drawn shows the mean relation which existed during the tests, between car weight and resistance at a definite speed. It is believed that the curves of Fig.10 are generally applicable to ordinary American freight trains, provided the conditions surrounding their operation are like those which prevailed during these tests.

[1] As has been previously explained, the curve is drawn by finding the "center of gravity" of several groups of points. These centers are defined in Figs. 3 to 9 by the crosses within circles. Points 34 and 74 were virtually ignored in drawing the curves of Figs. 6 and 7. The numbers at the points are the last two figures of the test numbers.

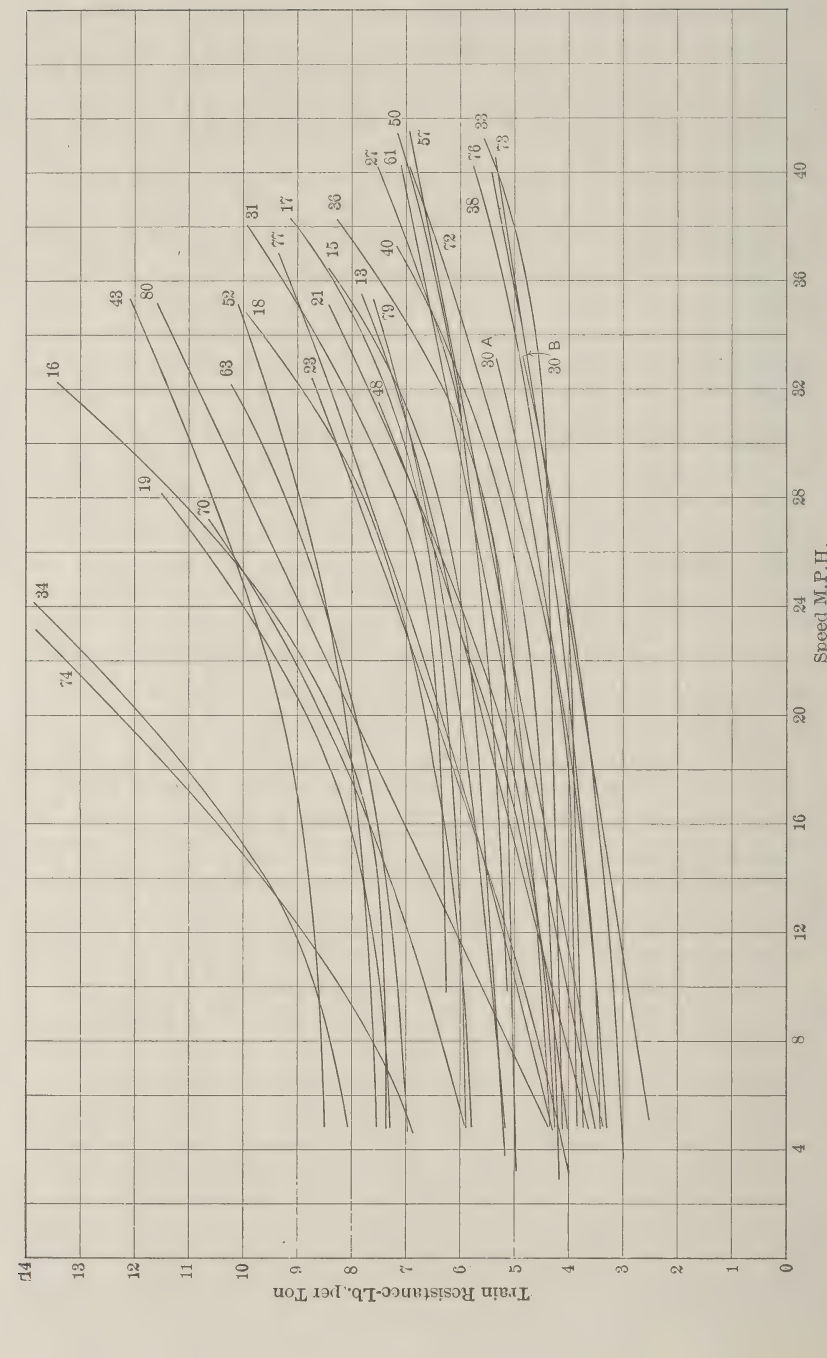

Fig. 2. Curves Showing Relation between Resistance and Speed for each of the 32 Tests

TABLE 2 RESISTANCE AT VARIOUS SPEEDS

DERIVED FROM THE CURVES FOR THE INDIVIDUAL TESTS. THIS TABLE PROVIDES THE COÖRDINATES OF THE POINTS PLOTTED IN FIGS. 3 TO 9

Test No.	Average Weight per Car Tons	Train Resistance—Pounds per Ton						
		5 m.p.h.	10 m.p.h.	15 m.p.h.	20 m.p.h.	25 m.p.h.	30 m.p.h.	35 m.p.h.
S-1016	16.12	7.35	7.40	7.62	8.37	9.91	12.22	.....
S-1034	16.56	8.10	8.70	9.92	11.90	14.30	.....	.....
S-1074	16.56	6.92	8.23	10.10	12.32	14.70	.....	
S-1043	16.92	8.50	8.61	8.85	9.30	10.00	10.95	12.04
S-1019	17.72	7.30	7.47	7.90	8.85	10.32	.....	.....
S-1063	20.04	6 98	7.13	7.43	7.90	8.63	9.63	....
S-1031	20.72	....	6.24	6.30	6.40	6.73	7.60	8.94
S-1080	21.40	4.40	5.57	6.75	7.94	9.15	10.35	11.55
S-1070	24.60	5.93	6.63	7.47	8.57	9.90	.....	.....
S-1052	24.80	7.55	7.63	7.80	8.10	8.55	9.20	10.05
S-1018	25.40	5.80	5.95	6.20	6.63	7.22	8.26	10.02
S-1077	28.40	4.32	4.91	5.58	6.34	7.15	8.01	8.96
S-1079	33.04	3.66	4.30	4.92	5.60	6.22	6 89	7.55
S-1015	36.08	5.20	5.36	5.52	5.70	6.02	6.71	7.95
S-1036	37.72	4.98	5.03	5.12	5.15	5.31	5.88	7.15
S-1013	38.04	....	5.40	5.65	5.95	6.32	6.90	7.68
S-1017	38.44	5.90	5.95	6.02	6.20	6.48	7.01	8.03
S-1023	38.72	4.16	4.80	5.56	6.40	7.30	8.25	....
S-1050	40.44	....	5.10	5.25	5.40	5.62	5.90	6.33
S-1057	41 32	3.40	3.88	4.35	4.83	5.31	5.80	6.30
S-1048	45.24	4.05	4.35	4.80	5.48	6.30	7.23	....
S-1040	45.76	4.22	4.30	4.40	4.58	4.90	5.52	6.53
S-1021	46.08	4.21	4.41	4.72	5.29	6.15	7.20	8.40
S-1027	47.44	4.31	4.48	4.67	4.90	5.22	5.79	6.55
S-1061	51.20	3.50	4.00	4.51	5.01	5.51	6.01	6.53
S-1033	51.72	4.10	4.15	4.20	4.25	4.32	4.40	4.65
S-1038	52.28	3.30	3.50	3.71	3.95	4.25	4.60	5.08
S-1030B	57.12	3.73	3.80	3.82	3.90	4.10	4.50	....
S-1030A	59.88	3.84	3.88	3.92	4.10	4.45	4.95	....
S-1072	66.40	3.40	3.50	3.70	4.10	4.61	5.27	6.00
S-1073	67.16	2.52	2.90	3.30	3.70	4.10	4.50	4.90
S-1076	69.92	2.97	3.13	3.37	3.70	4.04	4.49	4.95

The curves of Fig. 10 enable one to determine the probable mean resistance of any such train, at speeds between 5 and 35 miles per hour, provided the average weight of the cars composing the train be known.

70 *The Results Expressed as Resistance-Speed Curves.* While Fig. 10 presents the main results of the experiments, the form in which these results are there expressed is unusual. Ordinarily train resist-

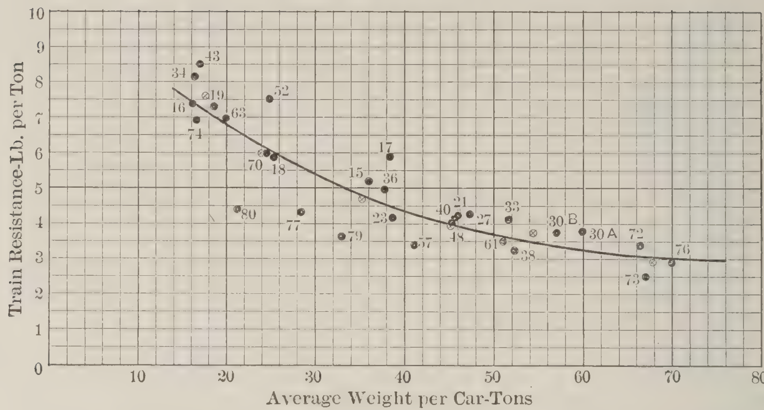

FIG. 3 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 5 MILES PER HOUR

ance is expressed either as a curve or equation which defines the relation between resistance and speed, instead of the relation between resistance and car weight as in Fig. 10. Obviously to express the results of these experiments in the usual form a single curve will not

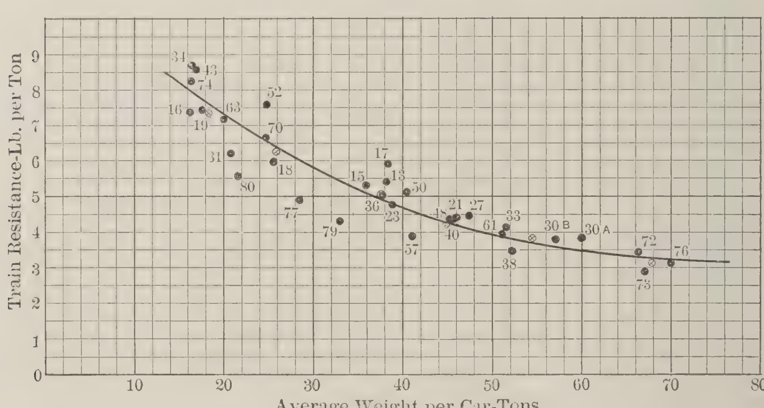

FIG. 4 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 10 MILES PER HOUR

suffice, since the influence of car weight cannot be thereby made evident. A number of curves will be required for this purpose, each of which will apply only to a definite average car weight. Fig. 11 presents such a group of resistance-speed curves, which have been

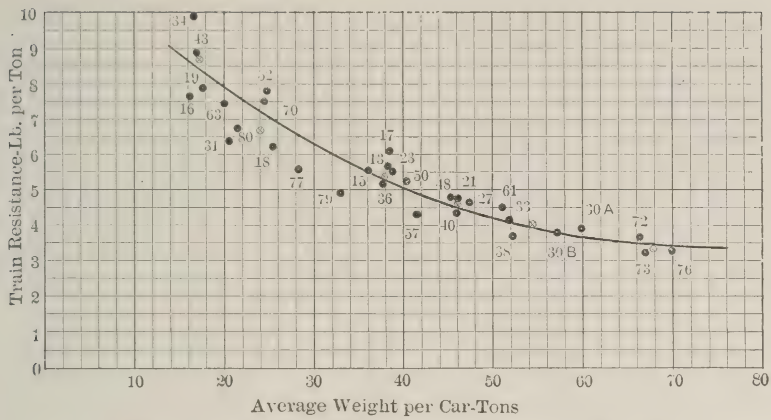

FIG. 5 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT

SPEED 15 MILES PER HOUR

derived directly from the curves of Fig. 10. Fig. 11 therefore exhibits in different form only such information as is obtainable from Fig. 10.

71 The relation between the two figures may be made clear by explaining the derivation of the upper curve in Fig. 11, the one apply-

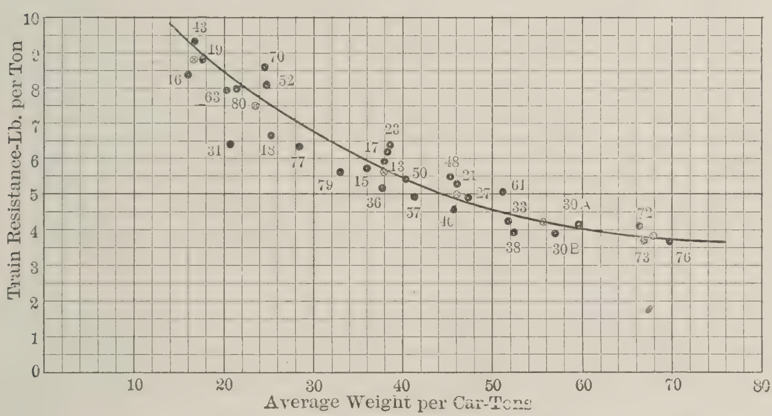

FIG. 6 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT

SPEED 20 MILES PER HOUR

ing to a car weight of 15 tons. In Fig. 10 the ordinate corresponding to an average car weight of 15 tons cuts the seven curves there drawn at seven points, at which the mean resistance values are 7.62, 8.20, 8.81, 9.56, 10.37, 11.24 and 12.25 lb. per ton, corresponding to

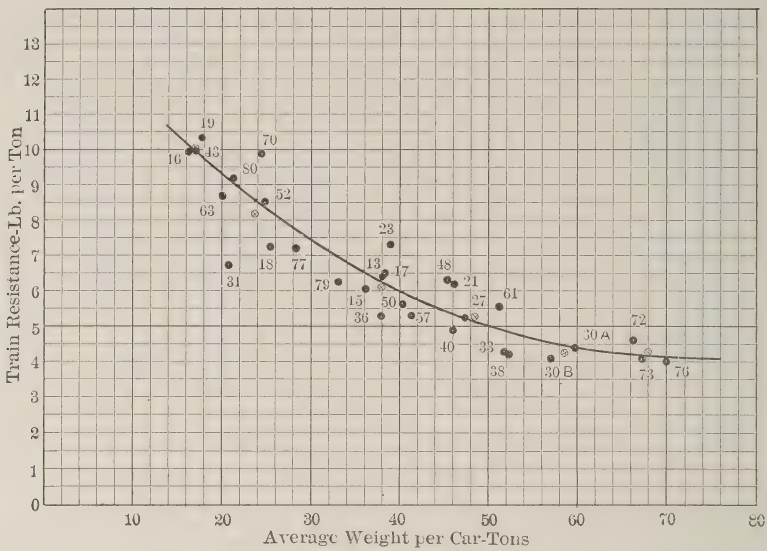

FIG. 7 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT

SPEED 25 MILES PER HOUR

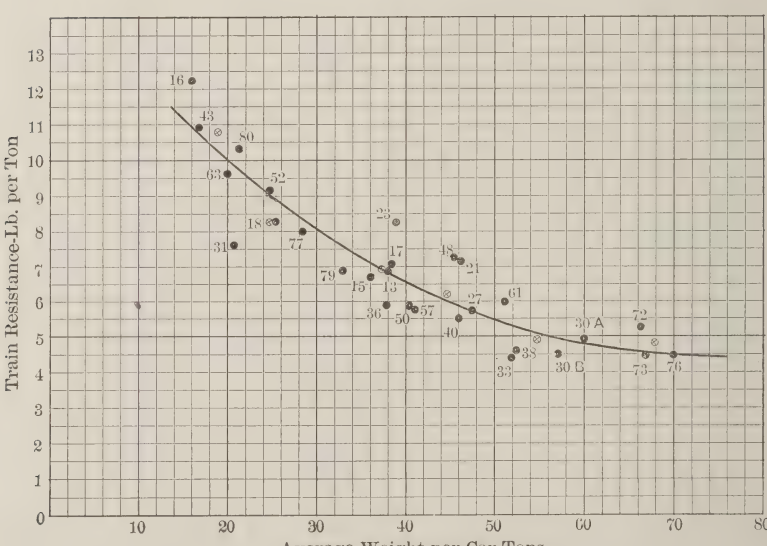

FIG. 8 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT

SPEED 30 MILES PER HOUR

speeds of 5, 10, 15, 20, 25, 30 and 35 miles per hour respectively These values are the coördinates of seven points on a resistance-speed curve applying to a car weight of 15 tons. These seven points have been plotted in Fig. 11, and the upper curve there shown has been passed through them and extended to 40 miles per hour. The other curves of Fig. 11 were derived by a like process. In the original diagram three additional curves, corresponding to 55, 65 and 70 tons per car, were drawn. These three curves have been omitted from the figure to avoid confusion. Fig. 11 reproduces quite exactly the facts presented in Fig. 10;* and presents the final results of the experiments.

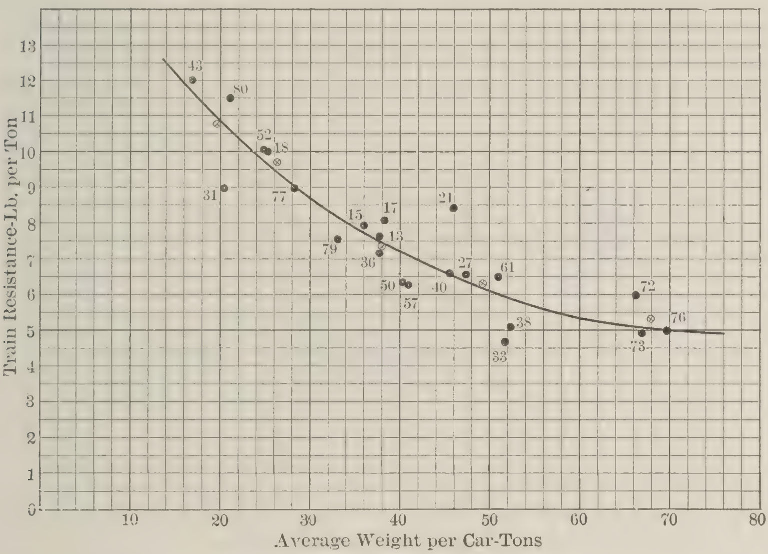

FIG. 9 RELATION BETWEEN RESISTANCE AND AVERAGE CAR WEIGHT
SPEED 35 MILES PER HOUR

72 *The Results Expressed in Tabular Form.* From each of the curves of Fig. 11 the values of resistance at various speeds have been determined and set down in Table 3. Table 3 also includes the coördinates of the resistance curves corresponding to 55, 65 and 70 tons per car, which are omitted from Fig. 11.

* The points derived from Fig. 10 have been omitted from the tracing from which Fig. 11 was reproduced. All such points lie very close to the curves drawn in Fig. 11, the maximum deviation amounting to but $\frac{3}{4}$ of one per cent of the corresponding curve ordinate. In the Appendix there are presented tables of coördinates, by means of which Figs. 10 and 11 may be exactly reproduced.

73 *The Results Expressed as Equations.* The relation between resistance and speed shown by each of the curves of Fig. 11 may be also expressed in the form of an equation. Formulæ 1 to 13 below are such equations, by means of which resistance may be calculated for any speed and for various car weights. In the formulæ, R is the resistance expressed in pounds per ton, S is the speed expressed in miles per hour, and W is the average weight of the cars in the train expressed in tons. The formulæ are purely empirical, and are simply equations of parabolas so selected as to correspond very closely with the curves of Fig. 11. The correspondence between the formulæ and

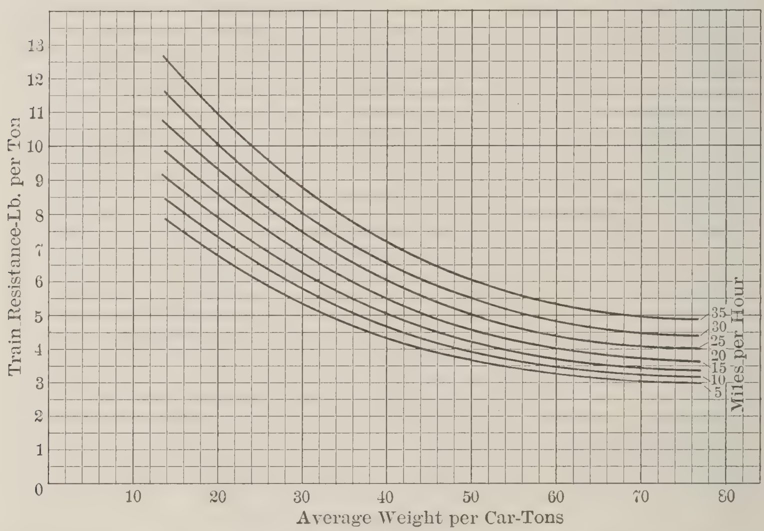

Fig. 10 Relation between Resistance and Average Car Weight at various Speeds

the curves is such that the maximum difference between any value of resistance obtained by the formulæ, and the corresponding value obtained from the curves of Fig. 11, is $\frac{1}{2}$ of one per cent. Since these are empirical equations their use should not be extended beyond the speed limits shown on Fig. 11.

74 It is possible to express approximately the facts presented in Fig. 11, in a single equation which includes only the first power of the three variables, R, S and W. Such an equation would obviously be more convenient than the group given below. Several such equations have been derived, each of which well represents, in general,

the results of the tests. Each of them, however, at some points in its range of application, leads to errors as great as 10 per cent. It has been deemed inadvisable to publish a formula containing so great an initial error.

75 *Final Results.* The final results of the research are presented in Fig. 11, in Table 3, and in Formulæ 1 to 13. It is believed that by means of either the figure, the table, or the formulæ, the resistance of ordinary freight trains may be fairly accurately predicted; provided the conditions surrounding their operation are similar to those which prevailed during these tests. These conditions have been fully stated and are restated in the conclusions. It is sufficient to repeat at this point that the results apply to trains running at uniform speed, on tangent and level track of good construction, during weather when the temperature is not lower than 30 deg. fahr. and when the wind velocity does not exceed about 20 miles per hour.

TRAIN RESISTANCE FORMULÆ

When $W = 15$ tons; $R = 7.15 + 0.085\ S + 0.00175\ S^2$ . . (1)
" $W = 20$ " $R = 6.30 + 0.087\ S + 0.00126\ S^2$ . . (2)
" $W = 25$ " $R = 5.60 + 0.077\ S + 0.00116\ S^2$ . . (3)
" $W = 30$ " $R = 5.02 + 0.066\ S + 0.00116\ S^2$ . . (4)
" $W = 35$ " $R = 4.49 + 0.060\ S + 0.00108\ S^2$ . . (5)
" $W = 40$ " $R = 4.15 + 0.041\ S + 0.00134\ S^2$ . . (6)
" $W = 45$ " $R = 3.82 + 0.031\ S + 0.00140\ S^2$ . . (7)
" $W = 50$ " $R = 3.56 + 0.024\ S + 0.00140\ S^2$ . . (8)
" $W = 55$ " $R = 3.38 + 0.016\ S + 0.00142\ S^2$ . . (9)
" $W = 60$ " $R = 3.19 + 0.016\ S + 0.00132\ S^2$ . . (10)
" $W = 65$ " $R = 3.06 + 0.014\ S + 0.00130\ S^2$ . . (11)
" $W = 70$ " $R = 2.92 + 0.021\ S + 0.00111\ S^2$ . . (12)
" $W = 75$ " $R = 2.87 + 0.019\ S + 0.00113\ S^2$ . . (13)

DISCUSSION OF THE RESULTS

76 *Variations in Resistance of Different Trains.* Reference has previously been made to the variations among the points of Figs. 3 to 9. In each figure about one-half of the points lie above the curve there drawn, and their resistance values vary from those of the curve by different amounts. It should be borne in mind that, in these figures each point represents the average resistance which prevailed throughout a particular test, and differences among the points represent, therefore, differences in the mean resistance of the different trains.

77 Among those trains which are regarded as normal there are two or three whose resistance at some speed varies from the mean, as expressed in the curves, by as much as 23 per cent. The great major-

ity, however, vary from this mean by about 10 per cent or less. In Fig. 4, for example, there are nineteen points which lie above the curve, among which the maximum deviation from the mean is 23 per

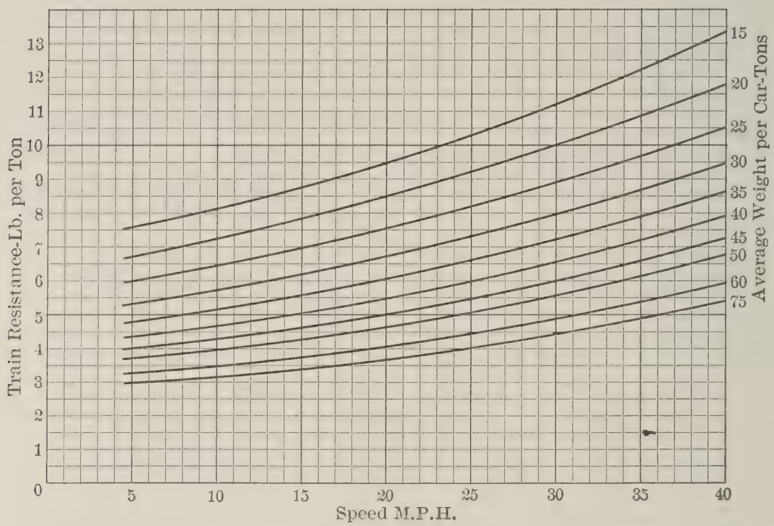

FIG. 11 RELATION BETWEEN RESISTANCE AND SPEED FOR VARIOUS AVERAGE WEIGHTS PER CAR

cent, while the average of the deviation for all nineteen points is 8 per cent. The following table presents similar average deviations above and below the mean for each of Figs. 3 to 9.

AVERAGE DEVIATION OF ALL POINTS IN FIGS. 3 TO 9, FROM THE MEAN AS SHOWN BY THE CURVES THERE DRAWN, EXPRESSED AS PERCENTAGES OF THE CURVE ORDINATES.

	Fig. 3 5 m.p.h.	Fig. 4 10 m.p.h.	Fig. 5 15 m.p.h.	Fig. 6 20 m.p.h.	Fig. 7 25 m.p.h.	Fig. 8 30 m.p.h.	Fig. 9 35 m.p.h.
Points above thecurve ...	11	8	8	11	13	8	7
Points below the curve..	13	10	9	8	9	9	9

78 The data present no satisfactory general explanation for these differences in the resistance of different trains of like average weight per car. They may be due to difference in external conditions, or in train condition and make-up. Whatever the explanation it is

TABLE 3 RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THE VALUES ARE DERIVED DIRECTLY FROM THE CURVES OF FIG. 11 AND REPRESENT THE FINAL RESULTS OF THE TESTS

Speed Miles per Hour	Train Resistance—Pounds per Ton													Speed Miles per Hour
	Column headings indicate the average car weights													
	15 tons	20 tons	25 tons	30 tons	35 tons	40 tons	45 tons	50 tons	55 tons	60 tons	65 tons	70 tons	75 tons	
5	7.6	6.8	6.0	5.4	4.8	4.4	4.0	3.7	3.5	3.3	3.2	3.1	3.0	5
6	7.7	6.9	6.1	5.5	4.9	4.4	4.1	3.8	3.5	3.3	3.2	3.1	3.0	6
7	7.8	7.0	6.2	5.5	5.0	4.5	4.1	3.8	3.6	3.4	3.2	3.1	3.1	7
8	8.0	7.1	6.3	5.6	5.0	4.6	4.2	3.9	3.6	3.4	3.3	3.2	3.1	8
9	8.1	7.2	6.4	5.7	5.1	4.6	4.2	3.9	3.6	3.4	3.3	3.2	3.1	9
10	8.2	7.3	6.5	5.8	5.2	4.7	4.3	4.0	3.7	3.5	3.3	3.2	3.2	10
11	8.3	7.4	6.6	5.9	5.3	4.8	4.3	4.0	3.7	3.5	3.4	3.3	3.2	11
12	8.4	7 5	6.7	6.0	5.4	4.8	4.4	4.0	3.8	3.6	3.4	3.3	3.3	12
13	8.6	7.6	6.8	6.1	5.5	4.9	4.5	4.1	3.8	3.6	3.5	3.4	3.3	13
14	8.7	7.8	6.9	6.2	5.5	5.0	4.5	4.2	3.9	3.7	3.5	3.4	3.4	14
15	8.8	7.9	7.0	6.3	5.6	5.1	4.6	4.2	3.9	3.7	3.6	3.5	3.4	15
16	9.0	8.0	7.1	6.4	5.7	5.1	4.7	4.3	4.0	3.8	3.6	3.5	3.5	16
17	9.1	8.1	7.2	6.5	5.8	5.2	4.8	4.4	4.1	3.9	3.7	3.6	3.5	17
18	9.3	8.3	7.4	6.6	5.9	5.3	4.8	4.5	4.1	3.9	3.7	3.7	3.6	18
19	9.4	8.4	7.5	6.7	6.0	5.4	4.9	4.5	4.2	4.0	3.8	3.7	3.6	19
20	9.6	8.5	7.6	6.8	6.1	5.5	5.0	4.6	4.3	4.0	3.9	3.8	3.7	20
21	9.7	8.7	7.7	6.9	6.2	5.6	5.1	4.7	4.3	4.1	3.9	3.9	3.8	21
22	9.9	8.8	7.9	7.0	6.3	5.7	5.2	4.8	4.4	4.2	4.0	3.9	3.8	22
23	10.0	9.0	8.0	7.1	6.4	5.8	5.3	4.9	4.5	4.3	4.1	4.0	3.9	23
24	10.2	9.1	8.1	7.3	6.6	5.9	5.4	4.9	4.6	4.3	4.2	4.1	4.0	24
25	10.4	9.3	8.3	7.4	6.7	6.0	5.5	5.0	4.7	4.4	4.2	4.1	4.0	25
26	10.5	9.4	8.4	7.5	6.8	6.1	5.6	5.1	4.8	4.5	4.3	4.2	4.1	26
27	10.7	9.6	8.5	7.7	6.9	6.2	5.7	5.2	4.8	4.6	4.4	4.3	4.2	27
28	10.9	9.7	8.7	7.8	7.0	6.3	5.8	5.3	4.9	4.7	4.5	4.4	4.3	28
29	11.1	9.9	8.8	7.9	7.1	6.5	5.9	5.4	5.0	4.8	4.6	4.5	4.4	29
30	11.3	10.0	9.0	8.0	7.3	6.6	6.0	5.5	5.1	4.9	4.7	4.5	4.5	30
31	11.4	10.2	9.1	8.2	7.4	6.7	6.1	5.6	5.2	5.0	4.8	4.6	4.5	31
32	11.6	10.4	9.3	8.3	7.5	6.8	6.2	5.8	5.3	5.0	4.9	4.7	4.6	32
33	11.8	10.5	9.4	8.5	7.6	7.0	6.3	5.9	5.4	5.2	5.0	4.8	4.7	33
34	12.0	10.7	9.6	8.6	7.8	7.1	6.5	6.0	5.5	5.3	5.1	4.9	4.8	34
35	12.3	10.9	9.7	8.8	7.9	7.2	6.6	6.1	5.7	5.4	5.2	5.0	4.9	35
36	12.5	11.1	9.9	8.9	8.0	7.4	6.7	6.2	5.8	5.5	5.3	5.1	5.0	36
37	12.7	11.2	10.0	9.0	8.2	7.5	6.9	6.4	5.9	5.6	5.4	5.2	5.1	37
38	12.9	11.4	10.2	9.2	8.3	7.6	7.0	6.5	6.0	5.7	5.5	5.3	5.2	38
39	13.1	11.6	10.4	9.4	8.5	7.8	7.1	6.6	6.2	5.8	5.6	5.4	5.3	39
40	13.4	11.8	10.6	9.5	8.6	7.9	7.3	6.8	6.3	6.0	5.7	5.6	5.5	40

significant that about one-half of the trains experimented upon developed a resistance about 9 per cent in excess of the mean resistance which would be predicted by the use of Figs. 3 to 9 and Figs. 10 and 11. Obviously a similar excess may be expected with any train; it is suggested therefore that in determining the resistance of trains on *level tangent track* for the purpose of rating locomotives under operating conditions which demand conservative ratings, 9 per cent be added to the resistance values obtained from the curves, tables and equations presented. Such considerations are of little practical importance in rating locomotives for speeds above 15 miles per hour. In such cases an excess in resistance over that expected can result in nothing more serious than failure to realize the expected train speed.

79 It should be understood that this 9 per cent allowance is intended to cover probable variations in the resistance of different trains under normal operating conditions. It in no way takes the place of that additional reserve which must be allowed to cover unusual variations in resistance due to low temperatures or high winds, nor of that reserve in the tractive effort of the locomotive necessitated by operating conditions which reduce the efficiency of the locomotive itself.

80 *Tests which Present Abnormal Resistance Values.* There are four points in Figs. 3 to 9 whose deviation from the curves is so great as to demand special examination. These are the points corresponding to Tests S-1034, S-1074, S-1080, and S-1031 (points 34, 74, 80, and 31). These tests show a persistent and great variation from the mean at various speeds. The trains of Tests 1034, 1074, and 1080 were like in average car weights, less than 23 tons, and in containing a large proportion of empty gondolas— 99, 98, and 84 per cent, respectively. Any explanation based on train composition is however nullified by the fact that the trains of Tests 1016, 1043, and 1063, which show close correspondence with the curves, had similar average car weights and contained almost equally large proportions of empty gondolas. Weather and wind conditions likewise offer no explanation of the divergences presented by these three tests. Explanations are rendered more difficult by the fact that while the trains of Tests 1034 and 1074 show unusually high resistance, the resistance in Test 1080 is exceptionally low. The abnormalities presented by these three trains have therefore been accepted as unexplained by the data at hand.

81 The resistance of the train of the fourth test mentioned above (S-1031) was low at all speeds. This train had an average car weight

of 20.7 tons, contained 94 per cent of box cars, and was only 1425 ft. long. Other test trains of similar average car weight differ generally in having less than 60 per cent of box cars, and all in being 2400 ft. or more in length. Taking into consideration all the data, neither fact seems to offer an adequate explanation, however, of the variations exhibited by this train.

82 *Car Weight as a Basis of Expression.* Objection may be made to the form of expression adopted in Figs. 3 to 9 and in Fig. 10, in which the resistance is expressed solely in terms of average car weight, to the apparent neglect of the influence of those elements of resistance, such as air resistance, which are independent of weight, and which probably vary only with the number of cars in the train. The neglect is only apparent, however, for the process by which Fig. 10 was derived involves, although indirectly, recognition of the influence of the number of cars. It is quite likely that if Fig. 10 were applied to determine the total resistance of a single car, the result would be in error.

83 Whatever objection may be urged against the form of expression adopted, it remains true that Fig. 10 rests upon experimental results obtained with trains of usual length and that in practice one is not likely to encounter trains which present in this respect any extreme variation from the test data. The form of expression will not lead to error unless misapplied, and it was chosen because the results may be conveniently used in establishing tonnage ratings.

84 It might likewise have been more rational to express the resistance in terms of load per axle instead of load per car, since the latter can operate to cause variations in resistance only in so far as it affects the former. Since, however, all American freight cars have four axles, the expression in either form would be identical. Convenience in application warrants the choice made in this respect also.

85 *Effect of Variety in Car Weight upon Total Train Resistance.* In Fig. 10 those portions of the curves which apply to average car weights below 20 tons were derived from trains which were quite homogeneous in their make-up as regards weight per car. These trains were necessarily composed almost exclusively of empty cars, since an average car weight of 20 tons or less cannot otherwise be obtained with cars of current design; and being empty they will be uniform in weight. Similarly for average car weights above 55 to 60 tons the test trains were necessarily uniform in make-up. For trains of average car weights below 20 and above 60 tons the curves of Fig. 10 are accepted, therefore, as valid, and applicable to any train to be met with in practice.

86 In Fig. 10, those portions of the curves corresponding to car weights of from 20 to 60 tons were, on the other hand, derived from trains which presented considerable diversity in make-up as regards weight per car. Some of these trains were composed almost entirely of loaded cars, others contained large proportions of both empty and loaded cars. In presenting the results in the form adopted in Fig. 10 (and Fig. 11), the assumption is that the curves there drawn will be used throughout their entire range of average car weight to determine the total resistance of both homogeneous and mixed trains, and that, when so applied, they will lead to no material error. In view of the facts just stated it is pertinent to inquire whether this assumption is justifiable.

87 Assume two trains of equal tonnage, and of the same average weight per car. Assume further that one is composed of cars uniform in weight, and the other of cars of different individual weights. Now if such trains are to have equal total resistance, it can be shown that the variation in resistance per car of the individual cars must be directly proportional to their weight. This implies that the curve showing the relation between total car resistance and car weight at a given speed must be a straight line, if homogeneous and mixed trains are to have equal total resistances at this speed. From Fig. 10 there have been derived such curves, showing the relation between car resistance and car weight. These curves (not shown in the report) correspond quite closely, but not exactly, with straight lines; and the correspondence is especially close for those portions of the curves which apply to car weights between 20 and 60 tons. From these facts we may conclude that the curves of Fig. 10 are not quite equally applicable to mixed and homogeneous trains, but nearly so, and that if the curves are applied to both kinds of trains, we may expect a slight error in the resulting total train resistance. The amount of such error is indicated by the following examination of a specific case.

88 Assume two trains, *A* and *B*, the first homogeneous, the second mixed, as regards car weight. Train *A* is composed of 60 cars each weighing 45 tons, and its total weight is 2700 tons. Train *B* is composed of 30 cars of 70 tons each and 30 cars of 20 tons each; its total weight is 2700 tons and its average car weight is 45 tons. Train *B* presents about as great a diversity in car weight as may be encountered in current practice. Both trains have equal tonnage and the same average weight per car. Assume that the total resistance of these two trains at a speed of 5 miles per hour is to be determined. By the procedure which it is intended shall usually be followed in using

Fig. 10, the resistance for an average car weight of 45 tons, at 5 miles per hour, is found to be 4.0 lb. per ton; and the total resistance of either train A or train B is $2700 \times 4.0 = 10{,}800$ lb.

89 Train B, however, may be considered as made up of two shorter homogeneous trains of average car weights of 20 and 70 tons respectively, and the resistance of each may be determined from those portions of the curves of Fig. 10 about whose validity no question is raised. From Fig. 10, the resistance at 5 miles per hour for a car weight of 20 tons is found to 6.8 lb. per ton, and for a car weight of 70 tons, 3.1 lb. per ton. By the use, therefore, of these portions of the curves of Fig. 10, the total resistance of train B is found to be $30 \times 20 \times 6.8 + 30 \times 70 \times 3.1 = 10{,}590$ lb., which differs from the resistance previously found by 2 per cent. If similar analysis be made for a speed of 40 miles per hour, the corresponding difference is found to be 4 per cent. If these differences be accepted as a measure of the maximum error likely to result from the indiscriminate application of the curves of Fig. 10 to mixed and homogeneous trains, we may conclude that for purposes of rating locomotives the results of the tests as expressed in Fig. 10 and 11 and Table 3 may be so applied without material error.

90 *The Influence of Speed on Resistance.* Within the last two years the opinion has been expressed in some quarters that train resistance between speeds of 5 and 35 miles per hour is constant. It is proper to point out that there is nothing in the data here presented to support such a conclusion.

91 *The Influence of Wind Velocity on Resistance.* The wind velocities prevailing during the tests were generally less than 20 miles per hour. The data do not permit the influence of such winds to be differentiated from the other elements affecting resistance; but they do warrant the conclusion that this influence is small. In the introduction, train resistance was defined as the resistance in still air, whereas throughout the report the term is applied to the test results from which the influence of wind has not been eliminated. This inconsistency has been deliberately incurred to avoid unwieldy expression, and is partially justified by the facts just stated.

92 *Comparison with other Experiments.* There is no point in comparing the results of these tests with formulæ in which the influence of car weight is given no consideration, nor with those not derived from tests on American cars of recent design. The results obtained on the Chicago, Burlington and Quincy Railroad and on the Pennsyl-

vania Railroad, and recently published by Mr. F. J. Cole[1] are selected for comparison.

93 The results obtained on the Chicago, Burlington and Quincy road ("curve No. 1, for temperatures above 30 deg. fahr. and no wind") apply to a speed of 20 miles per hour. Compared with the curve for 20 miles per hour in Fig. 10, they show resistance values from 35 to 60 per cent lower than the corresponding results of these tests. The Pennsylvania Railroad results are supposed to be equally applicable at all speeds between 5 and 30 miles. When plotted on Fig. 10 of this report they show very close correspondence with the curve there drawn for 10 miles per hour, for car weights from 25 to 70 tons; while for car weights below 25 tons they indicate resistance values as much as 20 per cent in excess of the results obtained during these tests.

[1] Railway Age Gazette, August 27 to October 1, 1909.

APPENDIX

RAILWAY TEST CAR NUMBER 17

The dynamometer car by means of which these tests were made was built in 1900. Under the arrangements then entered into, the car itself was built and has been since maintained by the Illinois Central Railroad; while the University has supplied all apparatus, and has manned and operated the car. Both the car body and the apparatus were remodeled in 1907.[1]

2 The car body was especially designed for its purpose. It is 40 ft. long over the end-sills, and 8 ft. 4 in. wide inside. The central sills and the platforms are of steel, while the remainder of the construction is of wood. The general design of the car is shown in Fig. 1, and an interior view is shown in Fig. 2. The working space occupies about two-thirds of the length of the car, and in it are placed the recording apparatus, the auxiliary instruments, the storage batteries, the work bench, etc.

3 During the tests which are here reported the test-car apparatus made continuous autographic records of drawbar pull, speed, time, mile-post positions, air-brake cylinder pressure, wind velocity with respect to the car, and wind direction with respect to the longitudinal axis of the car. These records were made upon a chart 36 in. wide, drawn across the table of the recording apparatus. During these tests this chart was driven by gearing from the axle of the central truck below the car, so that its travel was proportional to the travel of the car itself. A view of the recording apparatus is shown in Fig. 3.

4 Fig. 4 is reproduced from a tracing of a portion of the chart made during test S–1057 of this series. The only lines there shown which do not appear on the original record, are the profile and the transverse lines which mark the limits of one of the sections selected for calculation. These lines and some of the explanatory lettering have been added to the tracing, in order to make clearer the significance of the various records.

5 The total pull which comes upon the measuring drawbar of the car is transmitted to oil contained in the receiving cylinder, whose design is shown in Fig. 5. This cylinder is hung from the center sills, immediately behind the drawbar yoke. Its inside diameter is 10 in., and its piston is 7 1/2 in. long. Both cylinder and piston are carefully ground to an exact fit and no piston packing is used. The pull is transmitted from the drawbar yoke to the piston, through a roller-borne yoke; and the whole device is practically frictionless. Such leakage of oil as takes place proceeds so slowly as to prove of no inconvenience, even when operating under maximum pull. The cylinder may be refilled with oil

[1] A more detailed description of the present equipment is contained in an article by F. W. Marquis, which appeared in the Railway Age Gazette of February 19, 1909.

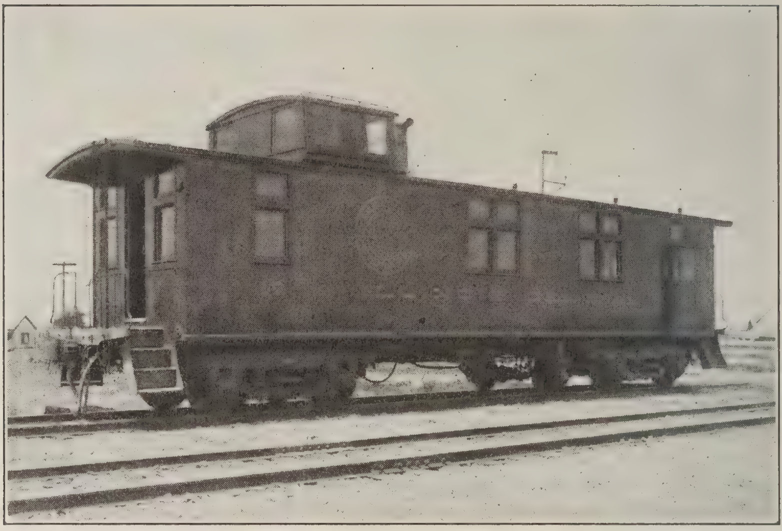

FIG. 1 RAILWAY TEST CAR NO. 17

FIG. 2 INTERIOR OF TEST CAR NO. 17

by means of a pump within the car while the car is in operation, without impairing the accuracy of the record. The pressure of the oil in this receiving cylinder is transmitted to the cylinder of an indicator located upon the table within the car. This indicator is identical in design with one of the modern types of steam-engine indicators; although it is larger and heavier throughout. During its ten years of service this dynamometer has demonstrated its reliability and accuracy.

6 Two speed records are shown on the chart, and both are used. The one is obtained from a speed recorder which resembles in design a "flyball" engine governor. This instrument is used in measuring speeds above 15 miles per hour. The second record is obtained from a chain-driven Boyer speed recorder, geared to run at about three times its usual speed. This record is used for speeds up to 35 miles per hour. Within their respective ranges, both instruments produce accurate speed curves.

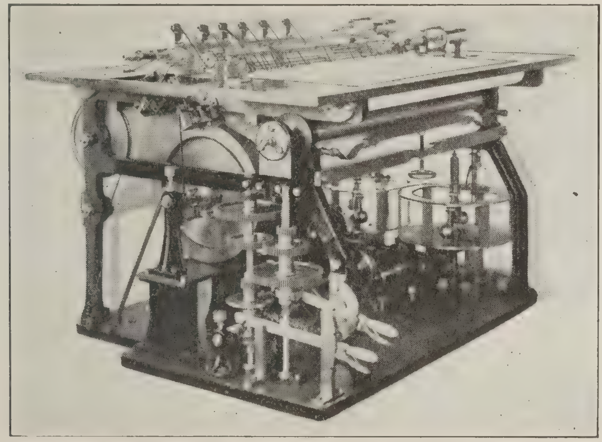

Fig. 3 The Recording Apparatus

7 The air-brake cylinder of the test car is connected to the cylinder of an ordinary steam engine indicator, which is mounted upon the table and which draws a curve of air-brake cylinder pressure.

8 The velocity of the wind with respect to the car is obtained by means of a Robinson cup-anemometer of the standard United States Weather Bureau type, which is so mounted that the cups revolve 32 in. above the car roof. This instrument controls an electric circuit, which operates an electro-magnet connected to the recording pen. By means of this magnet, offsets are made in the line drawn by the pen. During the time which elapses between two succeeding offsets, the actual movement of the wind amounts to 0.2 of a mile.

9 The direction of the wind with respect to the longitudinal axis of the car, is derived from a wind vane mounted 3 ft. above the car roof. The spindle of the vane extends downward to a point above the recording apparatus, and terminates there in a crank which is parallel to the vane. This crank is con-

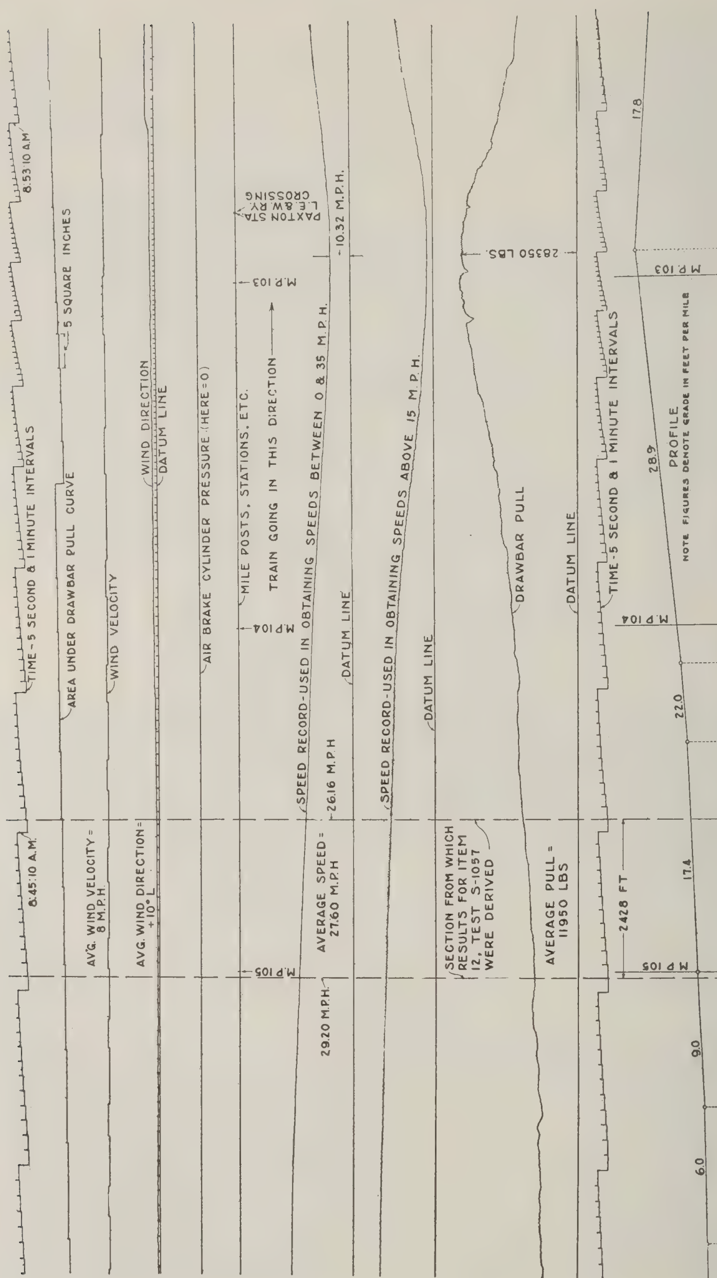

Fig. 4 A Portion of the Chart from Test S-1057

nected to the recording pen through a rod with a yoke end. The ordinate of the curve drawn by this pen is proportional to the sine of the angle made by the vane with the car axis. The offsets in the datum line for this curve which appear in Fig. 4, indicate that the vane, at the moment, was pointed toward the front end of the car.

10 Fig. 4 shows a record of "area under the curve of pull" which is made by means of a recording planimeter mounted on the table. This record is inaccurate and was not used in these calculations.

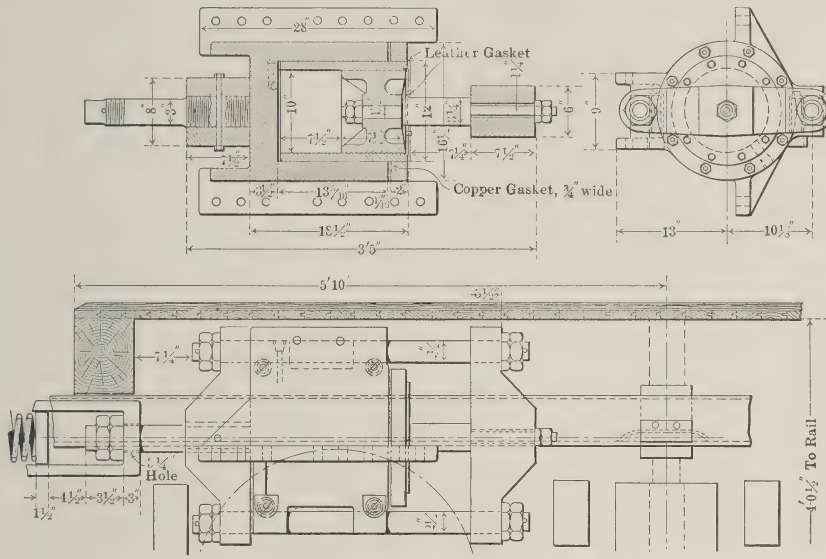

Fig. 5 Receiving Cylinder of the Dynamometer

EXACT COÖRDINATES FOR THE CURVES OF FIGS. 10 AND 11.

11 The original drawings from which Figs. 10 and 11 of the paper have been reproduced, were drawn to a scale about four times as great as that of the cuts. From these original drawings, the values of the coördinates of the various curves of both figures have been determined as accurately as possible; and these values are presented in Tables 1 and 2 herewith.

12 The curves of Fig. 10 (and of Figs. 3 to 9) of the paper may be accurately reproduced by the use of Table 4; and the curves of Fig. 11 may be reproduced from the values given in Table 2 herewith. The tables are presented merely to permit the accurate reproduction, to any scale, of the curves of the report; and are not intended for use in determining values of resistance. For the latter purpose Table 3 of the paper is more convenient and sufficiently accurate.

TABLE 1a RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THIS TABLE PRESENTS THE COÖRDINATES OF THE ORIGINAL CURVES FROM WHICH FIGS. 3 TO 9 AND FIG. 10 WERE REPRODUCED

Average Weight per Car Tons		Train Resistance—Pounds per Ton (column headings indicate the various speeds)							Average Weight per Car Tons	
		5 m.p.h.	10 m.p.h.	15 m.p.h.	20 m.p.h.	25 m.p.h.	30 m.p.h.	35 m.p.h.		
15		7.62	8.20	8.81	9.56	10.37	11.24	12.25		15
	16	7.44	8.00	8.61	9.34	10.13	10.98	11.95	16	
	18	7.10	7.63	8.22	8.92	9.68	10.47	11.39	18	
	20	6.77	7.30	7.85	8.53	9.26	10.00	10.89	20	
	22	6.45	6.97	7.49	8.16	8.84	9.56	10.41	22	
	24	6.16	6.64	7.14	7.79	8.46	9.16	9.94	24	
25		6.02	6.50	6.98	7.62	8.28	8.95	9.72		25
	26	5.88	6.35	6.81	7.44	8.10	8.77	9.52	26	
	28	5.61	6.07	6.51	7.11	7.76	8.40	9.12	28	
	30	5.38	5.80	6.23	6.80	7.43	8.05	8.75	30	
	32	5.13	5.54	5.98	6.51	7.12	7.72	8.40	32	
	34	4.92	5.31	5.72	6.24	6.82	7.40	8.06	34	
35		4.82	5.20	5.61	6.11	6.68	7.26	7.91		35
	36	4.72	5.10	5.50	5.99	6.55	7.11	7.77	36	
	38	4.55	4.90	5.28	5.74	6.29	6.83	7.48	38	
	40	4.38	4.70	5.06	5.50	6.03	6.57	7.20	40	
	42	4.22	4.52	4.88	5.29	5.80	6.32	6.95	42	
	44	4.08	4.38	4.70	5.09	5.59	6.10	6.71	44	
45		4.01	4.30	4.61	4.99	5.49	6.00	6.60		45
	46	3.95	4.21	4.52	4.90	5.38	5.90	6.49	46	
	48	3.82	4.08	4.38	4.71	5.20	5.71	6.28	48	
	50	3.72	3.96	4.24	4.56	5.03	5.52	6.10	50	
	52	3.61	3.85	4.11	4.42	4.88	5.36	5.91	52	
	54	3.52	3.75	3.99	4.30	4.74	5.20	5.74	54	
55		3.48	3.71	3.94	4.25	4.68	5.12	5.67		55
	56	3.43	3.67	3.90	4.20	4.62	5.05	5.60	56	
	58	3.37	3.58	3.81	4.10	4.50	4.93	5.47	58	
	60	3.30	3.50	3.73	4.02	4.42	4.83	5.36	60	
	62	3.23	3.44	3.67	3.97	4.34	4.74	5.27	62	
	64	3.18	3.39	3.60	3.90	4.29	4.68	5.18	64	
65		3.15	3.36	3.58	3.88	4.25	4.64	5.14		65
	66	3.12	3.32	3.55	3.85	4.22	4.61	5.11	66	
	68	3.09	3.30	3.50	3.80	4.18	4.57	5.06	68	
	70	3.05	3.26	3.47	3.76	4.13	4.52	5.01	70	
	72	3.02	3.22	3.44	3.73	4.10	4.49	4.98	72	
	74	3.01	3.19	3.42	3.71	4.08	4.48	4.93	74	
75		3.00	3.18	3.41	3.70	4.07	4.47	4.91		75

TABLE 2b RESISTANCE AT DIFFERENT SPEEDS AND FOR TRAINS OF VARIOUS AVERAGE CAR WEIGHTS

THIS TABLE PRESENTS THE COÖRDINATES OF THE ORIGINAL CURVES FROM WHICH FIG. 11 WAS REPRODUCED

Speed Miles per Hour	Train Resistance—Pounds per Ton													Speed Miles per Hour
	COLUMN HEADINGS INDICATE THE AVERAGE CAR WEIGHTS													
	15 tons	20 tons	25 tons	30 tons	35 tons	40 tons	45 tons	50 tons	55 tons	60 tons	65 tons	70 tons	75 tons	
5	7.62	6.77	6.02	5.38	4.82	4.39	4.01	3.72	3.49	3.30	3.16	3.05	3.00	5
6	7.73	6.86	6.12	5.46	4.90	4.43	4.07	3.77	3.52	3.33	3.19	3 08	3.03	6
7	7.83	6.97	6.21	5.53	4.98	4.50	4.12	3.81	3.56	3.37	3.23	3 12	3.07	7
8	7.96	7.06	6.31	5.62	5.04	4.57	4.18	3.86	3.60	3.40	3.26	3.16	3.10	8
9	8.07	7.18	6.40	5.71	5.11	4.62	4.22	3.90	3.64	3.44	3.30	3.20	3.13	9
10	8.19	7.29	6.50	5.80	5.20	4.69	4.28	3.96	3.69	3.49	3.34	3.24	3.18	10
11	8.30	7.40	6.60	5.90	5.29	4.76	4.33	4.00	3.73	3.52	3.38	3.29	3.21	11
12	8.42	7.51	6.71	5.98	5.37	4.83	4.40	4.04	3.78	3.58	3.42	3.33	3.26	12
13	8.56	7.63	6.81	6.08	5.46	4.90	4.47	4.11	3.83	3.62	3.47	3.38	3.31	13
14	8.70	7.76	6.92	6.18	5.53	4.98	4.53	4.18	3.89	3.68	3.52	3.43	3.36	14
15	8.82	7.88	7.01	6.28	5.64	5.06	4.60	4.24	3.94	3.73	3.57	3.48	3.41	15
16	8.98	8.00	7.12	6.39	5.73	5.13	4.68	4.31	4.00	3.80	3.62	3.53	3.47	16
17	9.10	8.13	7.24	6.49	5.82	5.23	4.75	4.38	4.05	3.86	3.68	3.60	3.52	17
18	9.25	8.27	7.37	6.60	5.92	5.32	4.83	4.45	4.12	3.92	3.74	3.66	3.58	18
19	9.40	8.40	7.49	6.71	6.01	5.41	4.91	4.52	4.19	3.98	3.81	3.72	3.64	19
20	9.56	8.53	7.60	6.82	6.11	5.50	5.00	4.60	4.27	4.04	3.88	3.79	3.71	20
21	9.71	8.69	7.72	6.93	6.22	5.60	5.08	4.69	4.32	4.11	3.94	3.85	3.78	21
22	9.88	8.82	7.86	7.03	6.33	5.70	5.17	4.78	4.41	4.18	4.00	3.92	3.84	22
23	10.02	8.97	7.99	7.14	6.44	5.80	5.27	4.86	4.49	4.25	4.07	3.99	3.92	23
24	10.20	9.11	8.11	7.27	6.55	5.90	5.37	4.94	4.58	4.33	4.15	4.06	3.98	24
25	10.37	9.26	8.25	7.40	6.67	6.01	5.46	5.03	4.66	4.41	4.23	4.13	4.04	25
26	10.52	9.42	8.38	7.52	6.79	6.11	5.57	5.12	4.75	4.50	4.31	4.21	4.12	26
27	10.71	9.57	8.51	7.65	6.91	6.21	5.67	5.22	4.83	4.58	4.40	4.29	4.20	27
28	10.89	9.72	8.67	7.78	7.01	6.33	5.78	5.32	4.92	4.67	4.48	4.38	4.29	28
29	11.06	9.89	8.81	7.91	7.12	6.45	5.88	5.43	5.01	4.76	4.57	4.46	4.36	29
30	11.25	10.03	8.96	8.04	7.26	6.58	5.99	5.53	5.11	4.86	4.66	4.53	4.45	30
31	11.43	10.20	9.10	8.18	7.39	6.71	6.10	5.64	5.21	4.95	4.75	4.63	4.53	31
32	11.63	10.37	9.26	8.31	7.51	6.83	6.21	5.76	5.32	5.04	4.85	4.73	4.62	32
33	11.84	10.53	9.41	8.46	7.63	6.96	6.33	5.87	5.43	5.15	4.95	4.83	4 72	33
34	12.04	10.71	9.57	8.60	7.78	7.08	6.47	5.99	5.54	5.26	5.05	4.92	4.82	34
35	12.25	10.89	9.72	8.75	7.91	7.20	6.60	6.10	5.67	5.36	5.16	5.01	4.92	35
36	12.47	11.07	9.89	8.90	8.04	7.35	6.73	6.23	5.78	5.48	5.27	5.12	5.01	36
37	12.69	11.23	10.04	9.04	8.19	7.49	6.87	6.36	5.90	5.59	5.38	5.22	5.12	37
38	12.91	11.42	10.21	9.20	8.33	7.64	7.00	6.49	6.02	5.71	5.48	5.33	5.22	38
39	13.12	11 61	10.39	9.36	8.48	7.79	7.13	6.63	6.15	5.83	5.60	5.44	5.33	39
40	13.35	11.80	10.55	9.51	8.62	7.93	7.29	6.78	6.28	5.95	5.72	5.55	5.45	40

THE STRENGTH OF PUNCH AND RIVETER FRAMES MADE OF CAST IRON

By Prof. A. Lewis Jenkins, Cincinnati, O.

Member of the Society

The actual stress relations existing in a simple cast-iron beam when subjected to a load have been the object of many investigations conducted by engineers during the past hundred years. These investigations consisted of numerous experiments and mathematical deductions for the determination of the elastic laws and physical constants involved in the analysis of stresses in straight cast-iron beams, but did not include the necessary deductions for the case of curved beams similar to punch and riveter frames.

2 The object of this article is to determine experimentally the relation between the ultimate strength of curved cast-iron specimens similar to punch and shear frames and the ultimate strength of the attached test bars and to compare the experimental results with those determined by the various methods of analysis used in designing castings of similar shape. In making this investigation many publications relating to the subject have been studied and the most important results many of which are only of historical interest are stated and discussed in the appendix.

FACTOR OF SAFETY

3 It has been contended that the ultimate strength of cast-iron machine parts is of no practical value since the working stress seldom exceeds 5000 lb. per square inch, and as Hook's law is sensibly true within this limit the ordinary formulas for beams should apply with a fair degree of accuracy. It is frequently desirable, however, to know the ultimate strength of such elements as punch frames in order to design the safety link which is supposed to break and save the frame in case of an accident or overload.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

4 The ordinary formulas assume that the stress in a beam is directly proportional to the load and factors of safety based on the ultimate tensile strength of the material are the same as if based on the ultimate strength of the beam. This is not true, however, for cast-iron because the stress is not directly proportional to the load. A beam 1 in. square and 12 in. between supports made of cast iron having an ultimate tensile strength of 20,000 lb. per square inch would support a load of 222 lb. midway between the supports with a factor of safety of 5 based on the ultimate tensile strength. Such a beam would probably fail, however, under a load in the middle of about 2000 lb. giving a factor of safety of 9 based on the breaking load of the beam. Hence, by using the ordinary beam formulas in designing straight cast-iron machine elements that are to be subjected to bending strains the actual working stress may be close to that desired, but the actual factor of safety based on the strength of the casting will be much greater than the factor of safety based on the tensile strength of the material.

Methods Employed in Making Tests

5 For many years engineers have recognized the value of experimental data on the strength of cast-iron beams and a considerable volume of literature on this subject is now available. There are, however, no published data on the strength of curved beams similar in shape to punch frames. The results of some experiments on the strength of crane hooks published in the October 7, 1909, issue of the American Machinist, seem to show that the Pearson-Andrews formula is true within the elastic limit for steel specimens of small throat depth. It will be noticed, however, that the value used for Poisson's ratio does not compare favorably with the values of this constant usually given for steel.

6 In view of the fact that there seems to be no rational method for determining the strength of curved cast-iron beams the writer believes that some experimental data on this subject may be of practical value and also serve as an incentive for further investigation by others.

7 Three forms of test specimens as shown in Figs. 1, 2 and 3 were used for these tests. These specimens were carefully designed with a view to having them of equal strength throughout, the calculations being based on the ordinary formulas for beams. To prevent any torsional moment due to eccentric loading the specimens were provided with two small hemispherical projections for receiving the load. A round and a rectangular test bar were cast with each specimen as

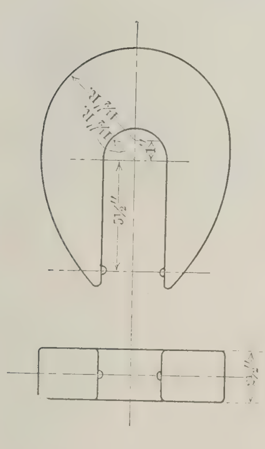

Fig. 1 Dimensions of Specimens 1, 2 and 3, Plate I

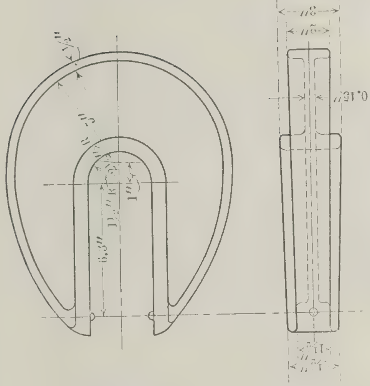

Fig. 2 Dimensions of Specimens 4, 5, 7 and 12, Plates I and II

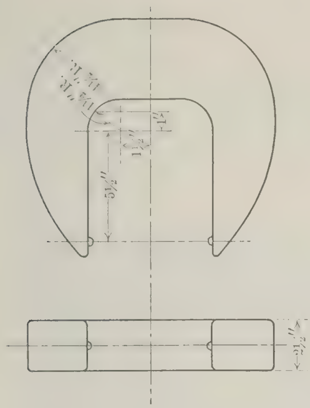

Fig. 3 Dimensions of Specimens 16 and 17, Plate III

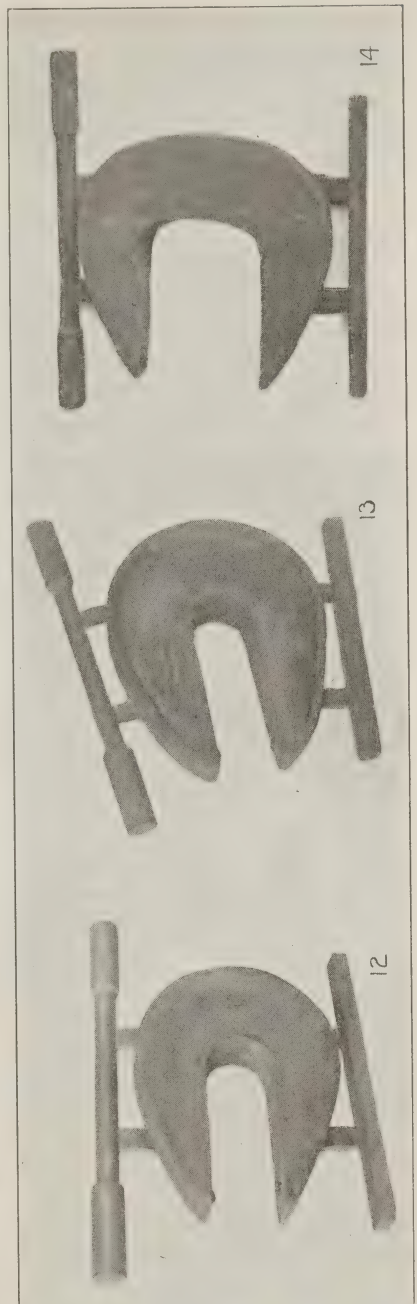

Fig. 4 Round and Rectangular Test Bars Cast with Each Specimen

shown in Fig. 4. The castings were made by the Buckeye Foundry Company of Cincinnati and poured from a heat run to cast lathe beds.

8 The 100,000 lb. Riéhle testing machine shown in Fig. 5 was used for making the tests. Fig. 6 shows the method employed for applying the load. Two stirrups forged from 1 in. by 3 in. steel and then tem-

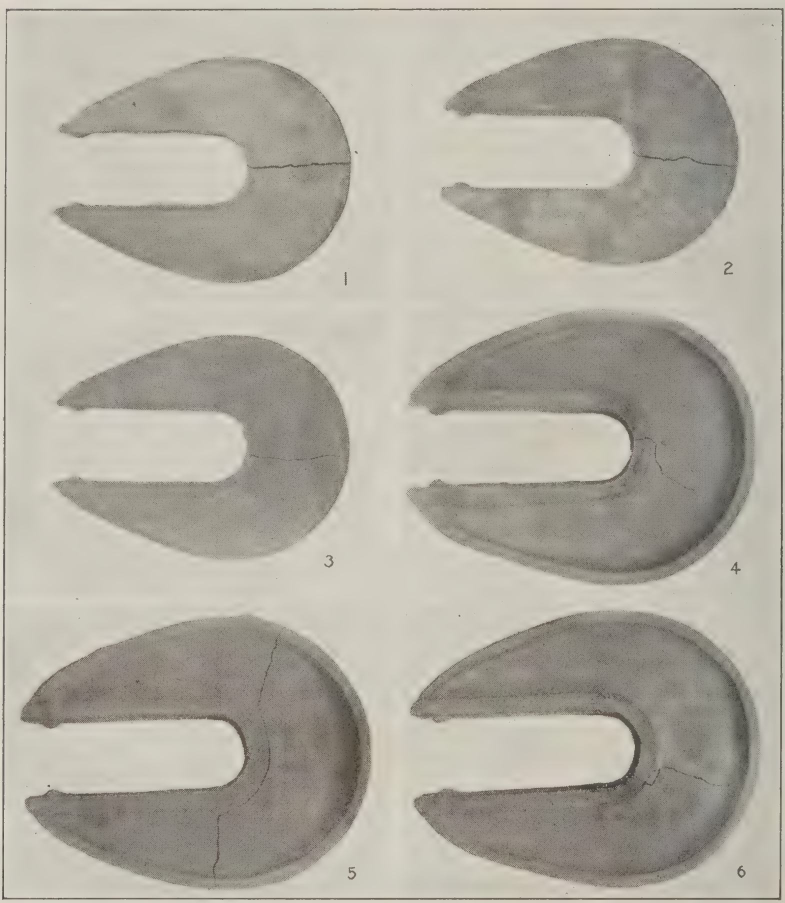

Plate I Curved Cast-Iron Specimens Broken by Tests

pered were held in the grips and received the castings. The tangs of the autographic recorder were placed between the casting and the stirrups, thereby eliminating the deflection of the stirrups and the bearing projections on the castings.

SIZE OF SPECIMENS

9 Plates I, II, and III were made from photographs of the broken specimens. The dimensions of specimens No. 1, 2 and 3 are shown in Fig. 1, and Fig. 2 shows the dimensions of specimens No. 4, 5, 7 and 12. The flanges on No. 12 were partially removed, but this did not

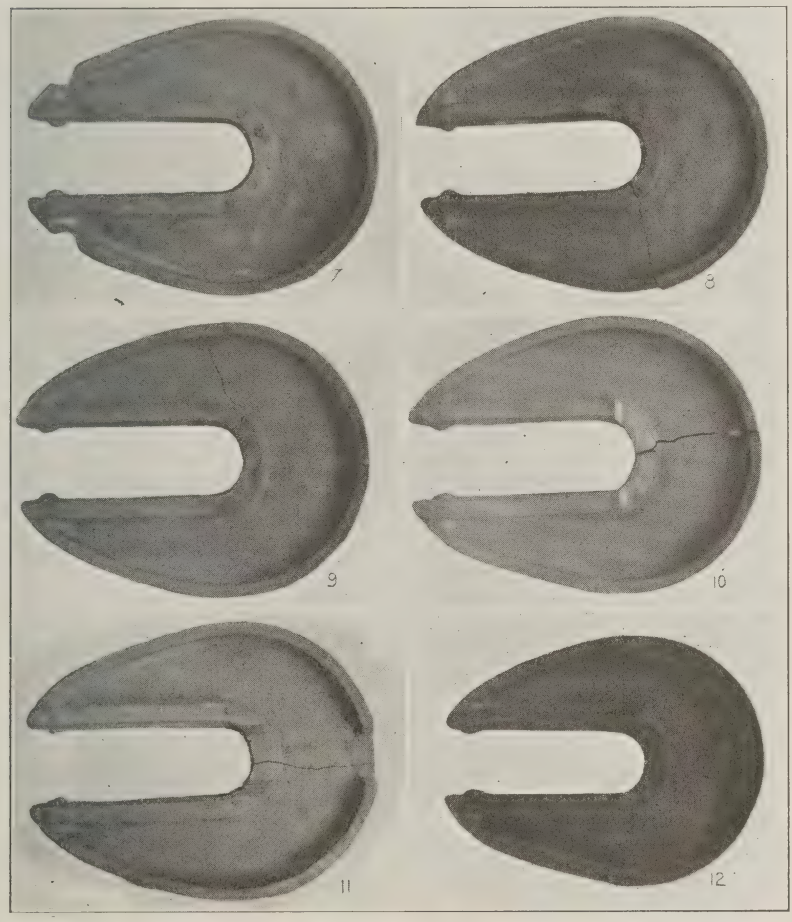

PLATE II. CURVED CAST-IRON SPECIMENS BROKEN BY TESTS

affect its strength. Specimens No. 9, 10, 11 and 13 were cast from the same pattern as No. 4 and the flanges altered as shown. No. 6 was cast from the same pattern as No. 4 after the thickness of the web had been increased to 0.93 in. No. 14 differs from No. 6 in that the fillet behind the flange is larger. No. 15 is the same as No. 14 with the out-

side flange removed. Fig. 3 shows the dimensions of No. 16 and 17. No. 18 is the same as No. 16 with depth of spine reduced to 3.4 in.

RESULTS OF TESTS

10 Table 1 gives the results of the tests. The unit tensile strength of the test bars varied from 18,600 to 24,400 lb., and the flexural

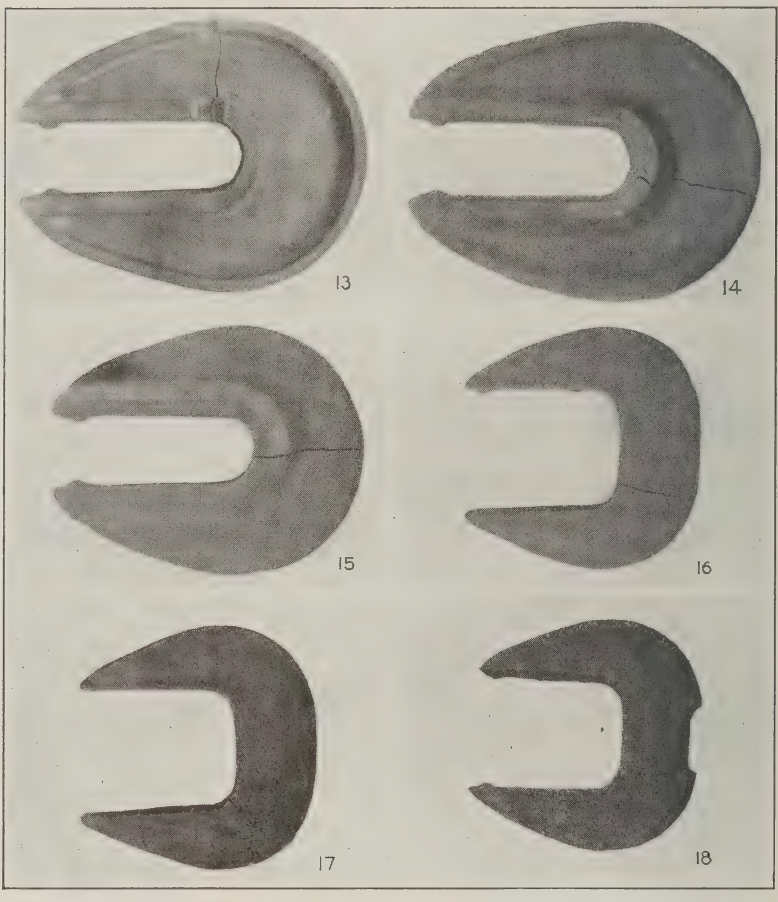

Plate III Curved Cast-Iron Specimens Broken by Tests

strength varied from 36,400 to 46,400 lb. The variations in the values of K seem to indicate that no definite relation exists between the tensile and flexural strength of small test bars.

11 The determination of the correct value for the radius of curvature of the gravity axis is extremely important and considerable care

was exercised in attempting to get accurate measurements of these values by plotting the curve of the gravity axis and finding its radius of curvature at the section considered in each case. It is possible for the personal equation to cause sufficient variations in these values to affect appreciably the final results. The chances for errors in plotting the transformed curves and measuring their areas is also worthy of the designer's consideration in choosing a formula.

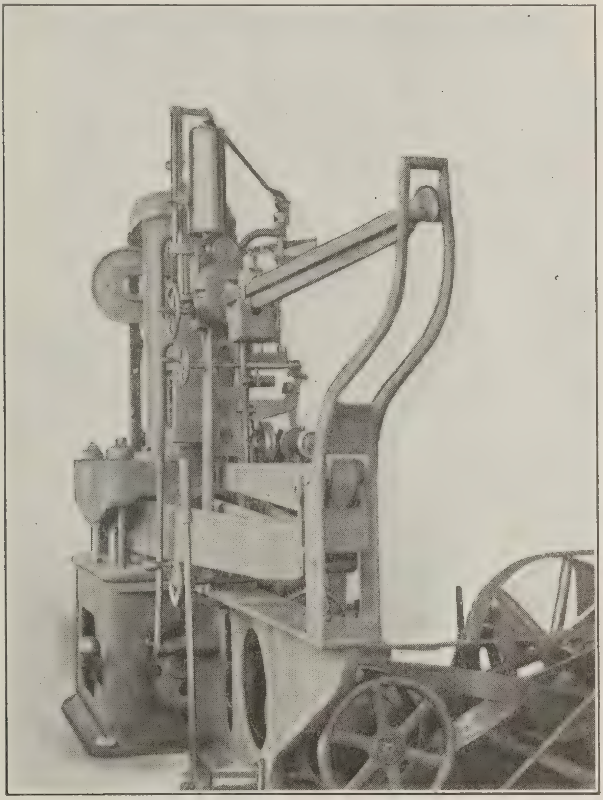

FIG. 5 100,000-LB. RIEHLE TESTING MACHINE

12 The average dimensions of castings No. 1, 2, and 3, Plate I, cast from the same pattern, are given in Fig. 1. Each casting was carefully measured and the values used in the respective calculations. The average tensile strength of the three test bars is 18,907 lb. per square inch. The maximum stress in the castings at failure as given by the beam formula is 14.2 per cent less than the tensile strength of the

test bar, whereas the stress according to the Résal and Pearson-Andrews formulas is 43.4 per cent and 75.4 per cent in excess of the strength of the test bar.

13 From Keep's experiments it is known that the unit strength of cast-iron bars of the same composition decreases as the area increases; hence, it is reasonable to suppose that the unit stress given by the beam formula is very close to the actual ultimate strength of the material.

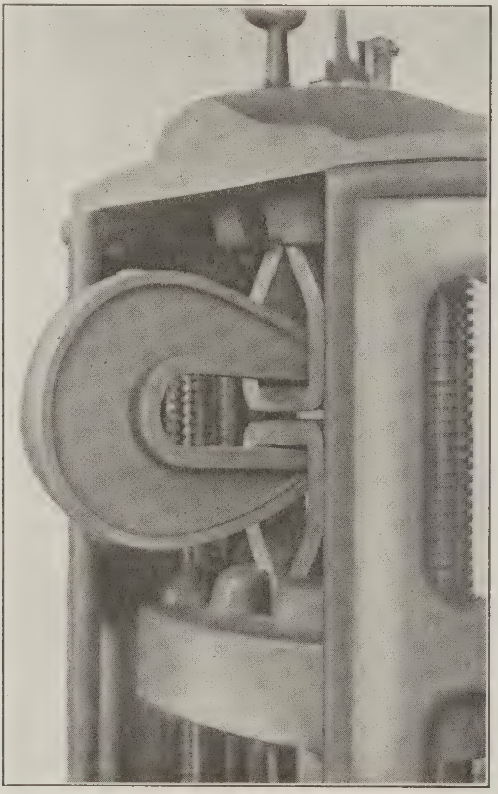

Fig. 6 Method Employed in Applying Load

The absurd results given by the Résal and Pearson-Andrews formulas in this case show that they are not applicable to this condition.

14 Castings No. 4, 5, 7, 9 and 12, Plates I and II, were cast from the same pattern and the average dimensions are given in Fig. 2. None of the above-mentioned formulas apply to these castings owing to the peculiar manner in which they failed. The results, however,

have an extremely important bearing on the design of punch frames. These specimens were supposed to fail under a load of about 15,000 lb. When the load on No. 4 reached 9300 lb. a fracture occurred behind the inner flange, as shown in the photographs of castings No. 7 and 12. Continued application of the load produced the fractures shown on No. 4, 5 and 9. The stress producing failure was evidently a normal stress as there is no shear on this section.

TABLE 1 RESULTS OF TESTS

Tension Specimens				Transverse Specimens Length 12 in.				Curved Specimens			
									Unit stress at ab		
Test No.	Breaking Load	Area Sq. In.	Stress Pounds Per Sq. In.	Breaking Load	Area (Breadth × Depth)	$S = \frac{MC}{I}$	$K = S_t \frac{bd^2}{M}$	Breaking Load	Beam Formula	Résal's Formula	Pearson-Andrews Formula
1	15000	0.785	19100	2380	.75 × 1.25	36560	3.14	11200	16240	27025	33013
2	14630	0.785	18620	2880	.75 × 1.25	44200	2.52	11125	16120	26844	32796
3	13460	0.709	19000	1680	1.25 × .75	46080	2.65	11390	16540	27484	33577
4	17000	0.785	21630	2280	1 × 1.05	37200	3.48	9300	11330	19583	38223
5	17000	0.785	21630	2000	1 × .95	40000	3.25	8500	10500	17909	34935
6	15500	0.833	18600	2160	1 × 1	39000	2.86	12600	22520	25870	29460
7	16300	0.866	18750	2380	1 × 1	43000	2.62	12000	9790	12584	27048
8	15070	0.724	21700	2310	1 × .95	46250	2.82	15300	12600	18420	
9	18000	0.785	22920	2200	1 × 1	39600	3.97	8300	10130	19164	36530
10	16000	0.785	20370	2300	.95 × 1	43700	2.70	8400	10520	——	——
11	17800	0.754	23600	2060	1.02 × 1	36400	3.94	5200	18420	21040	
12	18000	0.785	23000	2070	1 × .98	38000	3.56	8400	10235	17500	34524
13	18000	0.754	24400	2440	1 × .98	45000	3.20	5800	——	——	——
14	16300	0.739	21800	2300	1.02 × 1	40600	3.22	12700	23920	26435	28700
15	16100	0.754	21400	2240	1 × 1	40400	3.18	12500	23400	26025	28250
16	16730	0.785	21270	2460	.75 × 1.25	37800	3.41	11255	16320	27160	33300
17	17340	0.785	22080	2741	.75 × 1.25	42200	3.17	11980	17270	28987	35400
18	17900	0.785	22800	2680	.75 × 1.25	41300	3.35	10600	21476		

15 The conditions of stress that would produce such a failure may be given as follows:

a The inner flange being heavier than the web would tend to produce separation behind the flange due to unequal contraction when cooling. This initial stress due to cooling, plus the stress due to lateral contraction caused by the tensile stress due to bending, is a stress normal to the plane of

fracture. This method gives about 1500 lb. for the stress due to the load and about 18,000 lb. for the initial stress due to cooling, which is considered absurd. Hence this method does not offer a satisfactory explanation.

b In Fig. 7, given in the appendix, the load W produces a bending moment at the section AD equal to WN. This moment is resisted by the moment $\frac{SI}{c}$ where S is the sterss at A, I the moment of inertia of the section between A and D, and c the distance between the center of gravity of this section and the point A. The fracture of casting No. 5 suggested this method as a possible cause of failure. At a load of 8500 lb. a fracture occurred behind the inner flange and the load dropped almost to zero. Upon further application of the load the crack gradually approached the outer flange, then the inner flange fractured and finally the outer flange separated. According to this method of analysis the average unit stress at A in the five castings was about 22,700 lb. when the fracture occurred, which is excessive for tensile strength. This does not consider a possible shifting of the neutral axis, however, which would tend to decrease the value of the result. There are also reasons for using a larger value for N. By changing the values of N and c, it is possible to change the value for the stress, hence this method could not be relied upon without making many tests on different-sized specimens with different thicknesses of flange.

c Considering the flanges as separate members having pin connections at points A, E, F and G in Fig. 8, the load W would tend to increase the distance between the points A and G. If A and G be connected by means of a link, the stress in it may be found as follows: Draw the lines EG, FG, EA, and FA and let them represent links with pin connections. Then produce GA until it cuts the line of application of the load at K. The triangle AKE may be taken as the force diagram where $KE = \frac{W}{2}$ and KA is equal to half the stress in the link AG. In the specimen, this stress is taken by the web, the area of which is the outside

diameter, *pp*, of the inner flange multiplied by the thickness of the web. According to this method the unit stress is about 18,000 lb., which seems to express the stress relation very satisfactorily for this particular case, but it is not probable that such results would be derived from castings having a thicker flange.

16 Casting No. 7, Plate II, was first subjected to a compressive load acting on the horizontal side of the notches cut in the outer flange. This caused failure near the line of application of the load, as shown in the photograph. It was then subjected to a tensile load acting at a distance of $5\frac{3}{16}$ in. from the inside of the spine.

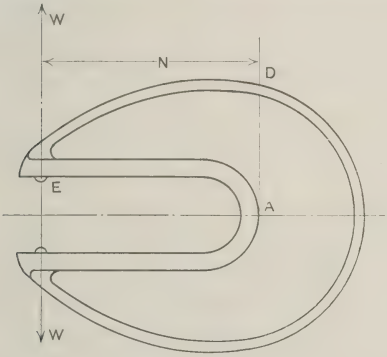

FIG. 7 DIAGRAM SHOWING CONDITION WITH BENDING MOMENT AT SECTION *AD*

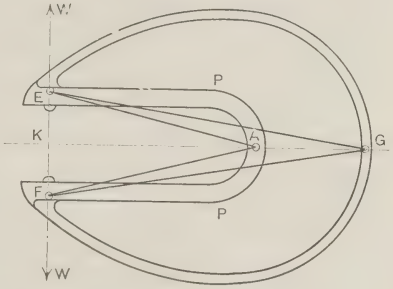

FIG. 8 DIAGRAM SHOWING CONDITION PRODUCING TENSILE STRESS IN SECTION *AG*

17 Casting No. 9 had $\frac{3}{16}$ in. of the outer flange removed on both sides. This, however, did not seem to affect its strength.

18 Casting No. 12 had the outer flange removed on both sides and the inner flange reduced to 1 in. wide at a distance of $6\frac{1}{2}$ in. from the load line; but this did not seem to affect its strength. The unit stress at the milled section according to the formula $S = \dfrac{Mc}{I}$ was 34,805 lb.

19 Casting No. 10 had the inner flange reduced to $2\frac{1}{2}$ in. wide and failed horizontally as shown. Failure in this manner was probably due to a flaw in the inner flange that extended into the web.

20 Casting No. 11 had about 2 in. of the outer flange removed on both sides and the width of the inner flange reduced to 2 in.

21 Casting No. 13, Plate III, had both flanges removed at a distance of 6.8 in. from the line of application of the load. The stress at

this point according to the formula $S = \frac{Mc}{I}$ is 31,160 lb. per square inch.

22 Casting No. 6 is the same as shown in Fig. 2 with the exception of having the thickness of the web increased to 0.93 in. The fracture occurred at an angle of about 30 deg. with the horizontal. There seemed to be a tendency for the fracture to follow the inner flange.

23 Casting No. 8 was subjected to a compressive load and failed at a section 7 in. from the line of application of the load, the stress being 35,370 lb. according to the formula for beams.

24 Casting No. 14 had the thickness of the web increased to 0.93 in. and provided with large triangular fillets behind the inner flange. By comparing the results of this test with those from No. 6, it seems that enlarging the fillet does not appreciably affect the strength of the casting.

25 Casting No. 15 is the same as No. 14 with the outer flange removed. The results seem to show that the outer flange on No. 14 adds but little to its strength.

26 Casting No. 16 failed in a curved portion. The angle between the plane of fracture and line of application of the load is about 79 deg. The stress in this section by Formula 7 (see Appendix) was 16,300.

27 Casting No. 17 failed in a curved portion, the angle being about 48 deg. The stress in the fractured section by Formula 7 (see Appendix) was 17,300.

28 Casting No. 18 is the same as No. 16 and No. 17 with the depth of spline reduced to 3.4 in. The stress at fracture was 21,476.

THE AUTOGRAPHIC RECORDER

29 The autographic recorder gave an apparently straight line for the deflection-load diagram for each specimen, but the scale of the curve is so small that a slight curvature due to the elastic law would not be perceptible. It is interesting to know, however, that the curve is so nearly a straight line and does not show any change in the elastic law which might be taken as the elastic limit of the specimen.

30 The load-deflection curve shows the total deflection produced by any given load on the specimen, and a sudden change in its direction would indicate that the stress in some section had reached the yield point. Any method of analysis involving the use of these curves is only applicable to the section which yields first. In the case of steel specimens it is difficult to determine this section, and in the analysis of stresses in cast-iron specimens the diagram finds no application.

CONCLUSIONS

31 Although these experiments are not sufficiently exhaustive to render any rigid conclusions, they seem to indicate that the following statements are approximately true:

a There is no rational method for predicting the strength of curved cast-iron beams suitable for punch and shear frames.

b Of the three formulas suggested for the design of punch frames, the well known beam formula,

$$S = \frac{Mc}{I} + \frac{W}{R}$$

is the most accurate statement of the law of stress relations existing in such specimens.

c The stress behind the inner flange at the curved portion is an important consideration that should be recognized by the designer.

d There seems to be no definite relation existing between the strength of a curved cast-iron beam and the transverse strength of a test bar cast with it.

e The Résal and Pearson-Andrews formulas are unwieldy and awkward in their application and offer many chances for error.

Acknowledgment is due to Prof. S. E. Slocum for criticisms during the preparation of this article and to Prof. John T. Faig whose assistance made possible the securing of the materials necessary for conducting the tests.

APPENDIX

ELASTIC LAW

Within the elastic limit, homogeneous materials such as steel and wrought iron conform with Hook's law, which states that the stress varies directly as the strain; hence the stress-strain diagram is a straight line. In common practice the elastic limit for cast iron is assumed to be about 6000 lb. per square inch and Hook's law is supposed to be practically true for stresses not exceeding this value but, as a matter of fact, cast iron has no definitely defined elastic limit like steel, and no portion of the stress-strain diagram is a straight line. The elastic law in this case may be expressed by the exponential equation

$$\Delta = K\,S^{m}$$

Where $\triangle$ denotes the unit deformation, S the unit stress, and K and m are constants. From the experiments of Bach these constants have been determined for cast iron, and the elastic laws found to be as follows:

$$\text{For tension, } \Delta = 0.00001111\,S^{1.0663}$$

$$\text{For compression, } \Delta = 0.00001444\,S^{1.0395}$$

2 These experimental curves may be replaced by parabolas without introducing any serious error. From experiments by Hodgkinson, the equations of the parabolas which fit these stress-strain diagrams most closely were found to take the form

$$S = 1{,}400{,}000\,\Delta\,(1 - 209\,\Delta) \text{ for tension}$$

$$S = 1{,}300{,}000\,\Delta\,(1 - 40\,\Delta) \text{ for compression}$$

3 By assuming that the equation of the elastic law is a parabola, the stress in a beam at a distance y from the neutral axis was expressed by St. Venant in the equations,

$$S = T\left[1 - \left(1 - \frac{y}{c}\right)^{m}\right] \text{ for tension}$$

$$S' = C\left[1 - \left(1 - \frac{y}{c'}\right)^{m}\right] \text{ for compression}$$

where T and C denote the ultimate tensile and compressive strengths, c and c' the distance from the neutral axis to the extreme fiber in tension and compression respectively, and m a constant so chosen as to make the parabola fit the stress-strain diagram for the given material. For the value $m = 1$, these equations reduce to the form

$$S = T\,\frac{y}{c}$$

which is Hook's law.

POSITION OF THE NEUTRAL AXIS AND THE MOMENT OF RESISTANCE

4 The equation

$$M = \frac{SI}{c}$$

is commonly known as the fundamental formula for beams. This formula is due to Navier and is based on the following assumptions: first, the strain is directly proportional to the distance from the neutral axis; second, the neutral axis coincides with the gravity axis; third, the stress is directly proportional to the strain on both the tension and compression sides. Some authorities favor the validity of the first assumption, but they all in general agree that the second and third are not true for cast iron.

5 The fact that the flexural strength given by the above formula does not coincide with the tensile strength has reflected discredit upon it. The flexural strength is from 1.5 to 2 times the tensile strength for ordinary test bars. The cause of this discrepancy is frequently explained by saying that the formula is based on Hook's law, and is not expected to hold true above the elastic limit, hence it does not give the actual tensile strength at rupture. This statement, however, is not necessarily true, because the formula will represent the stress relations in a rectangular beam under all conditions of load provided the coefficients of elasticity for tension and compression are equal.

6 The discrepancy is due to the shifting of the neutral axis which is caused by the difference in the elastic laws for tension and compression. A method for determining the position of the neutral axis is given by St. Venant in his notes on Navier's Resistance of Solid Bodies. Assuming that the strain varies directly as the distance from the neutral axis, the stress will vary as in the ordinary stress-strain diagram. Now if the beam fails in tension, the stress at rupture on the extreme fiber on the tension side is equal to the ultimate strength of the material.

7 Let $BE = S_t$ represent the ultimate tensile strength of the material, and let OE and ODF represent the stress-strain diagrams for tension and compression respectively as shown in Fig. 1. Then OB is the distance of the neutral axis from the tension side of the beam drawn to the same scale as yet undetermined, and the area OEB represents the total tensile stress in the beam at this section. But since the total tensile and compressive stresses must be equal, the compressive side of the beam is found by drawing a line AD such that the area OAD is equal to the area OEB. OA will then represent the distance of the compressive side of the beam from the neutral axis to the same scale as that to which OB is drawn. The scale may be easily determined by comparing AB with the actual depth of the beam.

8 The stress-strain diagram for the compression side is more nearly a straight line than that of the tension side. By assuming that the compression side obeys Hook's law, the formula for the moment of resistance is thereby simplified and becomes for a rectangular section of breadth b and depth d,

$$M = S\frac{bd^2}{6}\left[\frac{m\left(\frac{3(m+3)}{m+2} + 4\sqrt{\frac{2}{m+1}}\right)}{m+3+2\sqrt{2(m+1)}}\right]$$

The value of m in this equation is to be so chosen as to make the expression for the elastic law conform as closely as possible with the actual stress-strain diagram. This equation for any given material reduces to the form

$$S = \frac{MK}{bd^2}$$

9 Hodgkinson's[1] experiments led him to conclude that the neutral axis did not coincide with the gravity axis just at the point of failure of a cast-iron beam, that it moved toward the compression side, dividing the depth into a ratio of 1:5 or 1:6, which is the ratio of the ultimate strengths of the material in tension and compression.

10 W. H. Barlow[2] experimented with beams 7 ft. long, 6 in. deep and 2 in. wide, and found that loads less than three-fourths of the breaking load did not change the neutral axis materially, but just before rupture the shifting became greatly increased. These experiments accounted for only a small percentage of the discrepancy assumed by Hodgkinson.

11 Another explanation is that the outer fibers are subjected to an initial compressive stress due to their being cooled before the inner portion of the beam, and this initial stress must be overcome before the tensile stress begins.

12 Lewiston[3] proposed the theory that the compression side of a cast-iron beam is stressed to its ultimate strength at the point of failure. That is to say, a beam is stressed to its ultimate strength on both the compressive and tensile sides just before failure occurs. According to his theory, the distance from the neutral axis to the fiber that fails first is

$$X = \frac{I}{C + T} d$$

where C and T are the ultimate strengths in compression and tension and d the depth of the beam. As a result of this theory the moment of resistance of a rectangular cast-iron beam is

$$M = \frac{5}{18} S\, bd^2$$

when the ratio of the ultimate strengths, $\frac{C}{T}$, is equal to 5. This equation may also be written

$$S = \frac{MK}{bd^2}$$

where S denotes the ultimate strength of the material in tension.

13 Emery[4] claims that neither Hook's law nor Bernoulli's assumption are true for cast iron, that the medial portion between the neutral axis and the extreme fibers is stressed more than assumed by Bernoulli's assumption, which would necessarily relieve the outer fiber of a certain amount of stress. In devel-

[1] Hodgkinson: Experimental Researches on Cast Iron.
[2] Phil. Mag. 1855.
[3] Trans. Am. Soc. C. E., Vol. 35.
[4] Trans. A. S. M. E. Vol. 8.

oping a formula for the resistance of a rectangular cast-iron beam he assumes the elastic law,

$$\Delta = KS^m$$

for the tension side and Hook's law for the compressive side, which gives

$$M = S\, bd^2 \frac{\left(n + \frac{2\,m}{3}\sqrt{2\,m\,t}\right)}{\left(1 + \sqrt{2\,m\,t}\right)^2}$$

where S is the tensile strength of the material, n and m are constants from integration and t the ratio into which the neutral axis divides the depth of the beam. This equation may also be expressed

$$S = \frac{MK}{bd^2}$$

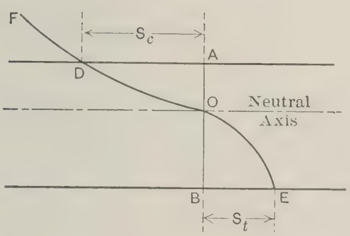

FIG. 1 STRESS-STRAIN DIAGRAM FOR TENSION AND COMPRESSION

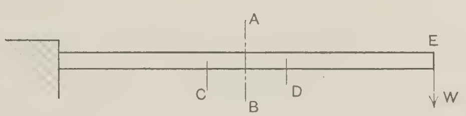

FIG. 2 CANTILEVER BEAM SUBJECTED TO A BENDING MOMENT

The values of K given by Mr. Emery range from 2.8 to 4.

14 Clark's formula,

$$S = \frac{WL}{1.55\, bd^2}$$

gives a value of $K = 3.46$.

15 Keep[5] published the results of a great many tests of different-sized specimens made of different grades of cast-iron and concludes that the silicon content and the rate of cooling greatly affects the strength of cast-iron. He also points out the fact that there is definite relation existing between the shrinkage and the percentage of silicon, and by measuring the shrinkage of a bar, the silicon content may be found from his shrinkage chart. This same chart is also provided with curves from which the strength of any bar between ½ in. and 4 in square may be found when the strength of any other bar within these limits is known. In the discussion of Mr. Keep's paper, Professor Benjamin proposed the formula

$$W = H \frac{b^{0.83}\, d^{1.89}}{L^{1.058}}$$

[5] Trans. A. S. M. E. Vol. 17.

where W is the breaking load and H a constant determined by breaking a bar 1 in. square and 12 in. long.

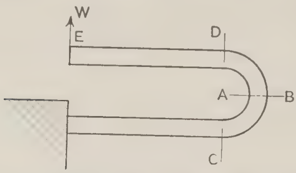

Fig. 3 Bent Cantilever Beam Subjected to a Bending Moment

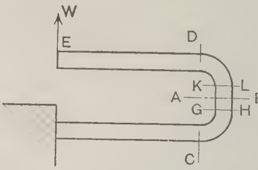

Fig. 4 Bent Cantilever Beam Subjected to a Bending Moment

16 Although based upon different theories and experiments, most of the above formulæ for straight cast-iron beams of rectangular cross-section reduce to the form

$$S = \frac{MK}{bd^2}$$

For rectangular sections the value of K has been found to vary between 2.8 and 4, depending upon the elastic laws of the material when subjected to tensile and

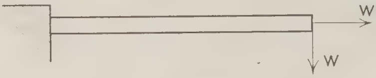

Fig. 5 Straight Beam Subjected to Bending Load

compressive stresses; and these laws vary with the sectional area, chemical composition, temperature of the metal when poured and the rate of cooling.

EFFECT OF CURVATURE AND FORMULAS FOR CURVED BEAMS

17 In Fig. 2 the simple cantilever beam is subjected to a bending moment due to the load W supported at the end. The bending moment at the section AB is resisted by the tensile and compressive stresses in the upper and lower fibers of the beam.

18 Consider the beam when bent into the forms shown in Fig. 3 and Fig. 4. The condition of stress in the portion ED is not affected by the change in shape; whereas the section AB resists not only the bending moment due to the load, but an additional force due to the direct pull of W.

19 The stresses in any section of a straight beam subjected to a bending load W, and a tensile load W' as shown in Fig. 5, may be accurately represented by the well known formulæ,

$$S_t = \frac{Mc}{I} + \frac{W'}{A}$$

$$S_c = \frac{Mc}{I} - \frac{W'}{A}$$

20 These formulæ are supposed to be true for straight portions of beams only; but due to their simple form, they are often applied to curved sections such as AB in Fig. 3; in which case $W' = W$ and $M = WL$, where L is the distance from line of application of the load to the gravity axis. When used in this way these formulas assume that the neutral axis coincides with the gravity axis of the section and its position is not affected by the curvature of the beam. These formulas may also be written in the form,

$$S_t = \frac{W}{A}\left(\frac{Lc}{R^2} + 1\right) \qquad [1]$$

$$S_c = \frac{W}{A}\left(\frac{Lc}{R^2} - 1\right)$$

where R is the radius of gyration and equal to

$$\sqrt{\frac{I}{A}}.$$

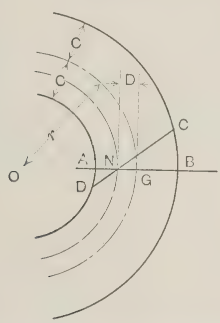

FIG 6 ENLARGEMENT OF CURVED PORTION FIG. 3

21 Fig. 6 shows an enlargement of the curved portion of the beam represented in Fig. 3. Assuming that a plane passed through the section AB before the load is applied remains a plane under the conditions of stress, that the stress is directly proportional to the distance from the neutral axis, and the radius of curvature of the gravity axis does not change; it will take less stress to deform the fibers on the convex side the same amount as on the concave side, because the convex side is longer. When the stresses at A and B are equal, the deformations may be represented by BC and AD respectively. The neutral axis does not coincide with the gravity axis at G, but passes through the point N which is nearer the convex or tension side, and at a distance D from G. The distance D, through which the neutral axis shifts, is independent of the stress when expressed by the equation.[6]

$$D = -\frac{\int \frac{y\,dA}{y + r}}{\int \frac{dA}{y + r}} \qquad [2]$$

[6] This method of analysis is due to Résal and is given in Strength of Materials by Slocum and Hancock.

where y denotes the distance from the gravity axis to any fiber having an area equal to dA, and r the radius of curvature OG of the gravity axis.

22 The stress on any fiber at a distance y from the gravity axis is expressed by the equation

$$S = \frac{M \quad (y + D)}{(y + r) \quad \int \frac{(y + D)^2}{y + r} dA} \qquad [3]$$

where the bending moment, $M = W (L - D)$.

23 The application of the above formula may be greatly simplified by a geometrical transformation of the section as shown in Fig. 7. Let $KLMN$ represent a section cut by the plane AB in Fig. 3, OS the axis of symmetry passing through the center of curvature O and GP the gravity axis of the section perpendicular to OS. By drawing radial lines from O through each point in the boundary, such as T, cutting the gravity axis at some point R, the lines drawn through T and R, perpendicular to OS and GP respectively will intersect in the point E. The locus of the point E is the boundary of the transformed curve $klmn$.

24 It has been proven by Résal that the distance between the center of gravity G of the original section and the center of gravity G' of the transformed section is equal to the value of D given in Equation 2, and the moment of inertia of the transformed section is equal to the integral in Equation 3. By denoting the moment of inertia of the transformed section by I', the above equation reduces to

$$S = \frac{Mr}{I'} \cdot \frac{y + D}{y + r} \qquad [4]$$

25 Another method of analysis is that due to Pearson and Andrews which takes into account the change in the radius of curvature and decrease in cross-sectional area due to lateral deformation. The tensile stress on the concave side is expressed by the formula

$$S = \frac{W}{A} \left[\frac{L}{r \gamma_2} \left(\frac{1}{\left(1 - \frac{c}{r}\right)^{n+1}} - \gamma_1 \right) + 1 \right] \qquad [5]$$

where

W = load in pounds;

A = area of cross-section in square inches;

L = distance from line of application of load to the gravity axis of the section;

r = radius of curvature of gravity axis in inches;

c = distance from gravity axis to extreme fiber on tension side;

n = Poisson's ratio of lateral contraction.

The value of γ_1 is determined graphically as follows: Let $KLMN$ in Fig. 8 represent the cross-sectional area cut by the plane AB in Fig. 3; r the radius of curvature, GP the gravity axis and y the distance from any point C on the boundary to the gravity axis. For each value of y, or position of D lay off

$$C\,E = \frac{C\,D}{\left(1 + \frac{y}{r}\right)^{n+1}}$$

forming the boundary *klmn*. The area of *klmn* divided by the area of the section *KLMN* is equal to the value of γ_1.

26 The value of γ_2 is equal to

$$\frac{\text{area of } klmn\text{—area } k'l'm'n'}{\text{area of } KLMN}$$

where the boundary of the area $k'l'm'n'$ is formed by making

$$CJ = \frac{CD}{\left(1 + \frac{y}{r}\right)^n}$$

for each value of y.

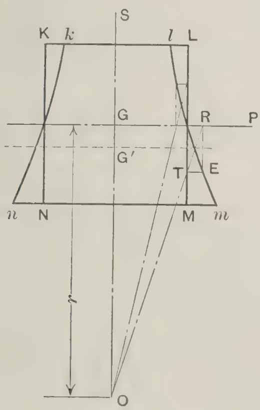

Fig. 7 Deformation of Section *AB* of Fig. 3 when under Stress

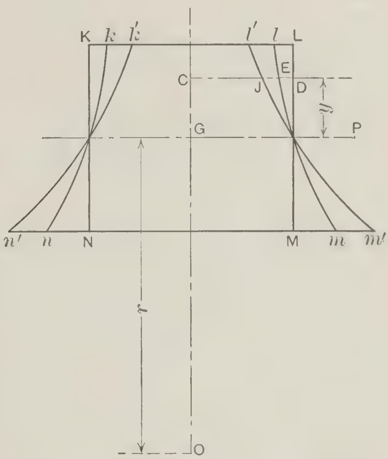

Fig. 8 Deformation of Section, under Pearson and Andrews Theory

27 The stress in any section of the straight portion of the beam shown in Fig. 3 may be found from the formula

$$S = \frac{Mc}{I}$$

and the stress in a section perpendicular to the line of application of the load, such as *AB*, may be found from Formula 4, provided the beam is of homogeneous material and obeys Hook's law, but none of the above formulas can be accurately applied to the section between *A* and *C*.

28 It is not necessary, however, to investigate more than one section of the curved portion provided the throat depth is small, as in the case of hooks; but in designing large punch frames it is desirable to determine at least two sections within the curved portion.

29 By comparing the properties of cast iron with the assumptions upon which the Formulas 1, 3, and 4 are based, it is easily seen that they do not apply to cast iron with any degree of accuracy. Hodgkinson's experiments

seem to show that due to the properties of the material, the neutral axis of a cast-iron beam shifts toward the compression side a distance

$$D' = \frac{d}{3}$$

Résal's formula as given in Equation 2 when applied to a rectangular section shows that

$$D = r - \frac{d}{\log_e \dfrac{2\,r + d}{2\,r - d}}$$

30 If the radius of curvature be such as to make $D' = D$, it seems that Formula 1 would apply. By equating the values of D and D' and solving, $r = 0.0505\,d$, which is a value that could not be used in practice. Hence, there is apparently no rational formula that represents the relation between the break ing load and the unit strength of curved portions of cast-iron beams similar in form to that shown in Fig. 3.

31 Press and punch frames requiring a wide gap are usually shaped similarly to the beam shown in Fig. 4, in which case a portion of the spine is straight. The sections KL and GH may be considered as belonging to either the straight or curved portions. If considered as belonging to the curved portion formula 4 should apply; but if taken as belonging to the straight portion it is subjected to the same conditions of stress as the section AB, and formula 1 should be applicable. These formulas, however, give different results which seem to indicate that the section AB should be less than the sections KL and GH of the curved portions, hence the condition of stress at KL and GH is somewhat similar to that due to sudden change in cross section, or which exists in the section connecting a spherical end to a thick cylinder.

32 A formula for determining the ultimate strength of castings having straight spines should recognize the change of the neutral axis due to the elastic properties of cast-iron. Such a formula based on the knowledge of straight beams of rectangular section takes the form

$$S = \frac{MK}{bd^2} + \frac{W}{A} \qquad [6]$$

where K is a constant to be determined experimentally.

33 The stress at sections between A and D in Fig. 3 and in sections above KL in Fig. 4 are usually determined by the formula

$$S = \frac{Mc}{1} + \frac{W \sin \theta}{A} \qquad [7]$$

where M is the product of the load into the distance between the line of application of the load and the gravity axis of the section considered; and θ the angle between the line of application of the load and the section produced until they intersect. For rectangular sections this formula may also be written

$$S = \frac{MK}{bd^2} + \frac{W \sin \theta}{A} \qquad [8]$$

FIRES: EFFECTS ON BUILDING MATERIAL AND PERMANENT ELIMINATION

By Frank B. Gilbreth, New York

Member of the Society

A remarkable paper, Bulletin 418, just issued by the United States Geological Survey, entitled The fire tax and waste of structural materialism in the United States, by Herbert M. Wilson and John L. Cochrane, contains the following astounding data:

The total cost of fires in the United States in 1907 amounted to almost one-half the cost of new buildings constructed in the country for the year. The total cost of the fires, excluding that of forest fires and marine losses, but including excess cost of fire protection due to bad construction, and excess premiums over insurance paid, amounted to over $456, 485, 000, a tax on the people exceeding the total value of the gold, silver, copper, and petroleum, produced in the United States in that year.

The actual fire losses due to the destruction of buildings and their contents amounted to $215,084,709, a per capita loss for the United States of $2.51. The per capita losses in the cities of the six leading European countries amounted to but 33 cents,or about one-eighth of the per capita loss sustained in the United States. In addition to this waste of wealth and natural resources, 1449 persons were killed and 5654 were injured in fires.

2 In discussing this waste, Charles Whiting Baker, editor of Engineering News, in an address before the meeting of the national engineering societies on the Conservation of Natural Resources, March 24, 1909, said:

The buildings consumed, if placed on lots of 65 ft. frontage, would line both sides of a street extending from New York to Chicago. A person journeying along this street of desolation would pass in every thousand feet a ruin from which an injured person was taken. At every three-quarters of a mile in this journey he would encounter the charred remains of a human being who had been burned to death.

The American Society of Mechanical Engineers, 29 W. 39th Street, New York. All papers are subject to revision.

The results obtained indicate that the total annual cost of fires in the United States, if buildings were as nearly fireproof as in Europe, would be $90,000,000, and therefore that the United States is paying annually a preventable tax of more than $366,000,000, or nearly enough to build a Panama canal each year.

It will be incidentally noted that fire protection involves the use of 2,000,000 tons of metal, having a value in excess of $127,000,000, and the metal in 350,000 hydrants, having a value of $30,000,000, all of which is wasted on account of the need of preparing to fight fires of a kind which, because of the inflammable character of building construction in this country, would develop into conflagrations without adequate water service and fire departments.

The estimated cost of private fire protection, including capital invested in construction and equipment, aggregates about $50,000,000, and the annual interest on this sum and the annual cost of watchmen's services amount to about $18,000,000.

3 In this age of "conservation" it is amazing that so little has been done to prevent destruction by fire, and to apply the lessons that every great fire teaches us. The interest of the entire public in fireproof construction flares up with a great fire, but flickers out before the building permits for new structures are granted. The average owner when rebuilding even finds fault with the authorities whose duty it is to enforce the inadequate existing ordinances relating to noncombustible construction and to "fire-stopping" in combustible buildings.

4 There must probably be more big fires among buildings now built, but there is no sane excuse for any such loss in the buildings to be built in the future. Within the last six years there have been wonderful opportunities for studying great fires here in our own country. It has been possible to observe the destruction that occurred, to discover why it occurred, and to determine exactly what must be done if repetition of such destruction is to be prevented in the future.

5 The writer has been fortunate in being able to make careful observation of the great fires at Toronto, Sioux City, Baltimore, San Francisco, Chelsea and elsewhere. Having become interested in the effects of fires as observed in Sioux City and Baltimore, the observations in San Francisco were made with great care and detail. The ruins of each of the 503 city blocks in the San Francisco fire were visited and studied. This led to the formulation of definite conclusions which have since been again checked by the Chelsea fire. The conclusions here presented, therefore, are not the result of theorizing or reading only, but of painstaking and directed observation.

6 All great fires are alike. Building material behaves the same

in a fire regardless of the location. We will therefore study a typical building in a typical fire.

7 The building of the Mutual Life Insurance Company at San Francisco was a steel frame structure, eight stories high, of the best construction in 1892, when it was erected. The laying up and filling of the joints in the brick, stone and terra cotta were as nearly perfect as possible. The exterior wall completely enclosed the steel frame, which was put together with bolted connections. The floors were of hollow terra cotta flat arches, and the partitions were hollow terra cotta blocks. The damage to the building, which necessitated the removal of the upper six stories, was practically all done by fire.

8 This building is excellent for an illustration, because it shows the good and bad points of many different kinds of incombustible materials, which were used in its construction. We took it down after the fire, photographing every part as it was taken down.

9 Examinations while taking down a building give the ideal method of studying the cause and effect and possible remedy for damage by fire.

10 The pictures illustrate better than could any description the process and the stages of removal. They show also that a very small quantity of wood in a so-called "fireproof" building built almost entirely of non-combustible materials will furnish sufficient heat to destroy it. The lessons from this and from all fires point to these conclusions: *No structure of the future should be built of wood. No structure of the future should contain any wood.* It has taken costly lessons to teach us this, but if we have at last learned the lesson, the price we have paid is cheap.

11 It has long been realized that, with the wood supply of the world constantly diminishing, wood was bound to become too expensive to use as a building material. But it is not so widely realized that the day when wood is as costly as non-combustible building materials is *here.* Today, concrete structures, and in some localities other non-combustible structures, can be erected at no greater first cost than wooden ones; with an added element of safety to those who use the building which cannot easily be overestimated.

Cost of Concrete Construction

12 In order to hold the place it claims—that of the cheapest and best fireproof construction material—concrete must be able to show itself cheapest in first cost, as well as cheapest in the long run.

13 Concrete materials are obtainable everywhere: stone, ledges, gravel, sand, burnt ballast, slag, brick refuse, terra cotta chips, stone cutters' refuse and cinders, are all good aggregates for concrete.

14 As far as the cost of the labor of putting the material in place is concerned, much has been done toward reducing that to a minimum. With metal forms and certified concrete, concrete construction is cheaper today than that with brick, hollow terra cotta tile or any other material.

15 Charles T. Main, a member of the Society, has recently published the most valuable contribution regarding cost data of brick and wood mill construction. Mr. Main is a leading authority on this subject, and his data are conceded to be correct. Mills have been built entirely of concrete, in all parts of the country, in less time than it is possible to build them of brick and wood and for less money than the square foot and cubic foot prices collected by Mr. Main. A few years ago this statement would not have been true. Today, with the aid of new designs, new methods, new forms, scientific grading and proportioning, it is a fact that cannot be disputed, but one, nevertheless, not generally known.

16 The development of the entire concrete building and of the parts of the concrete building have been slow, because it has taken a long time to learn to *think* in concrete. Each student brought his stock of experiences from another field to bear on the problem, but the problem was so new that few could understand it in all its aspects. The engineer, who knew how to design the strength, knew little about artistic features. The architect, who alone could give beauty to the building, was not an expert in the methods that make low costs. Progress was made more difficult by the fact that so few of the structural features and construction methods were standardized.

17 The contractor, having analyzed his costs, saw that the forms were the largest single item in the cost of the concrete. This fact had its effect on the design of the concrete building, in the trend toward straightway repetition form work and flat-slab ceilings, with as little beam and girder work as possible. A further benefit was in the fewer corners for fire to attack or for interruption of the action of sprinklers, saving also in the cost of the sprinkler system.

18 After the flat-slab ribless type of ceiling, with its low cost of forms, came standardization of the units of forms for walls and columns. With the use of these metal form-units a new problem faced us, namely, slight modification of design to agree with the cheapest labor requirements of the metal forms. The influence was

also felt, of the fire protection design thoroughly worked out and standardized by Charles L. Norton, professor of heat measurements at the Massachusetts Institute of Technology and engineer in charge of the engineering experiment station of the Factory Mutual Fire Insurance Companies. Mr. Norton's recommended practice, which is now universally accepted, is so simple and apparently so obvious that we sometimes forget to give him the credit for his work.

19 The importance of the forms, from the standpoint of economy, in being able to build fireproof buildings for the price of brick and wooden mill construction, is clearly shown by the fact that Charles D. Watson, of Syracuse, N. Y., on account of the reduction in the number and expense of forms, has been able to build a number of buildings from separately cast members and reduce the costs below anything ever cast *in situ* in wooden forms.

USE OF METAL FORMS

20 We come now to the latest developments, the metal form-units that have been perfected, densifiers for compacting the concrete, mechanical conveyors for displacing the wheel-barrow and hand tamping, and the concrete-producing factory, centrally located. These things have all arrived. Metal molds have reduced the price of concrete to that of wooden construction.

21 For the greatest saving in the cost per cubic foot of the completed building, the methods of construction, as well as the designs, must be standardized. This is rapidly being done. For example, the metal forms are set up and concrete is hauled in large quantities from a centrally located factory and poured at once for the entire story of a building. This does away with the joints between the work of two different days, and permits of figuring the tensile strength of the concrete in design calculation, with a great consequent saving of reinforcement. Such calculations cannot be safely figured unless the whole story is poured at once.

BEARING ON ULTIMATE COST OF HEALTH CONSIDERATIONS

22 Figuring the cost of furring, lathing, plastering, interest, fixed charges, heating, and repairs, concrete is certainly cheaper than wood. If you also figure health, concrete is *very much* cheaper. A concrete building is cooler in summer and warmer in winter than any other kind. It can be cleaned out with a hose. It can be easily, quickly and thoroughly fumigated, room by room, or all at once. It is drier

than any other, provided it is built with self-ventilating air spaces in all of the exterior walls. A concrete building is vermin-proof, and should germs or vermin get in the building, it may be easily rid of them.

23 The 1907 Year Book of the Department of Agriculture says (Page 98):

No one of our wild mammals, possibly not all combined, does as much damage as the common rat. . . . .

The rat continues to cause great losses throughout the United States. During the past year an attempt was made to ascertain the approximate damage done to property by this rodent, in the cities of Washington and Baltimore. Many business men were interviewed, including dealers in various kinds of merchandise, feeders of horses, managers of hotels and restaurants, and manufacturers. The inquiries included all sections of the two cities, and both small and large dealers. Data were obtained from some six hundred firms and individuals, from which it was estimated that the annual loss from rats in Washington is about $400,000; in Baltimore, upwards of $700,000. Assuming, as is probable, that similar conditions obtain in all our cities of over 100,000 inhabitants, the damage by rats in these centers of population entails a direct loss of $20,000,000 annually. This enormous sum gives an idea of the still greater total loss inflicted by this rodent throughout the length and breadth of the land.

The rat continues also to excite grave apprehension, because of its agency in distributing the dreaded plague and other diseases. Boards of Health and the Marine Hospital Service in several of our maritime cities have been prosecuting active war against the rodents, and large sums have been expended in efforts to effect their extirpation. No one method has proved adequate, and only by concerted, systematic, and persistent efforts is it possible to reduce and keep down their numbers. The rat-proof construction of buildings, the constant use of traps, and the use of poisons wherever possible, will go far toward assuring public safety. Experiments with various poisons and mechanical means of destruction have been made during the year, and a report of the subject with recommendations will soon be issued.

24 If wooden buildings are to be tolerated because of their alleged low cost, consider then that the concrete structure furnishes no hiding places for the rats and mice that cause damage amounting to a large rate of interest on over a billion dollars annually. There is no danger in exterminating rats by wholesale poisoning in concrete buildings, for there are no cracks for them to crawl into and die.

Advantage of Concrete Construction in Elimination of Fire Loss

25 It may seem that we have wandered from our subject of fires and their permanent elimination, but it was necessary to show that concrete is thus, from *every* standpoint, cheaper than wood, its

only rival for low cost. Concrete construction is the best form for the elimination of fires, because it affords:

a Least cause for fires.
b Least amount of damage to structural parts by fire.
c Least amount of damage to structural parts by water.
d Least amount of damage to contents of building by water.
e Least quantity of combustible structural materials in a room.
f Least speed of combustion of the contents of a room.
g No concealed fires; all fires are in plain sight.
h Least spread of fire to adjoining parts of the same story.
i Least spread of fire to stories above.
j Least spread of fire to next buildings.

26 The causes of fires this paper will not discuss. The amount of damage done by a fire in a concrete building depends upon circumstances perfectly within our control and predeterminable. With concrete made of properly selected fire-resisting materials, practically no damage is done, except by prolonged high temperature. The results of recent tests by Prof. Ira H. Woolson, a member of the the Society, and his assistant, J. S. Macgregor, prove conclusively that a concrete building properly designed, with as few projecting corners as possible, will withstand long periods of the hottest hardwood fires, with no resulting damage that cannot be thoroughly repaired with mortar and a plasterer's trowel. These tests were on full-sized rooms with walls of concrete made of different kinds of material.

27 Concrete for walls can be poured in metal molds with sufficient accuracy to permit of painting or wall-papering without further plastering or smoothing. This means that the best of this fire-resisting material is brought to the very surface of the wall where the flames strike. With metal molds all corners can be made rounded as cheaply as square, or metal corner beads may, if desired, be inserted and retained in the molds. Now, if a fire does occur in a building made of concrete cast in smooth metal forms, the damage is less than in any other type of building and the danger of spreading is less.

28 A concrete building can be damaged by water less than any other form of construction. To begin with, water does not injure concrete; in fact it improves its quality. There is no wood to swell, and afterwards to shrink and crack the plastering. Even the paint best suited to be used on concrete lends itself particularly well to

washing down with water. There are no hollow spaces that the water can flow through, damaging the contents below. A concrete building is water-tight from floor to ceiling, and small quantities of water can be easily handled through small scuppers, either into the air space of the vaulted wall, or through the wall to the outside. The fire is never hidden by the construction; consequently no unnecessary streams of water are flooded into the building.

29 In a concrete structure there need be little or no combustible material. Let us take for example the dwelling house, for it is the most difficult to make both cheap and fire-proof. In a concrete residence there is little trim that cannot be made better and cheaper of portland cement than of wood. The chair rails and picture molding can be made of concrete. The trim around the windows and doors can be molded in metal molds as cheaply as straight members. Even the wire moldings can be done away with, and the conduits buried in the concrete partitions, walls, ceilings and floors. Baseboards should be made of concrete or else omitted entirely—as they serve no useful purpose in a concrete building, except in following wooden precedents. Windows may have cement sashes, with wired glass, and self-closing shutters, or self-dropping shutters of rolled-up metal or asbestos. Metal furniture may be used. The paint and varnish used on buildings and furniture should be selected carefully, as these are great factors in determining the temperature at which a fire will start and the speed with which it will spread. There is also a great difference in the paints and varnish used for painting concrete.

30 The flooring need not be of wood. There are many first-class non-combustible materials beside portland cement that will fill every good requirement of wood and still be fireproof. There are also parquetry floorings, made of slow-burning wood, and much thinner than the old-fashioned hardwood floor, which make the best flooring in case wooden floors are desired. In a concrete residence there is no excuse for wooden under-flooring nor for wooden screeds.

31 The best form of construction for the elimination of fires is one that keeps the fire in plain sight, once it is started. One of the greatest obstacles in fighting fire in wood construction is the difficulty of locating a fire after it is known to exist. Every second of time lost gives it accelerated fury. Holes cut through the walls, floors and ceilings to locate a fire oftentimes only furnish more drafts to feed the flames.

32 With hollow concrete blocks, or monolithic cast-in-place walls

and partitions there are no chances for the fire to be concealed. There is no cause for cutting into the partitions, floors or ceilings. There is no damage to the structure from cutting, no injury from water. With such a building, if a fire does start, it starts in plain sight remains in plain sight at all times and can be attacked instantly before it has spread; and the water damage will be practically nothing. Fires in plain sight are usually extinguished before the arrival of the fire department.

Checking the Spread of Fires

33 The spread of a fire into adjoining parts of the same story is possible in a concrete building only through doorways, pipe holes, etc. A concrete wall is an ideal barrier to the spread of any fire. Not only is it incombustible, but it is unaffected for a long period by any but extremely high temperatures. Its coefficient of expansion, 0.0000055 to 0.0000060, is practically the same as that of steel; consequently, there is no damage from unequal expansion of the concrete and its steel reinforcement. As for doorways, any of the makes of fireproof doors and windows well made in accordance with the requirements of the Boston Mutual Fire Insurance Company will stop completely almost any fire. There are metal-covered doors made that very few people can distinguish without touching from mahogany or other hard woods. These doors will confine any ordinary fire to one room in a concrete building; or they will hold any fire in one room long enough to enable the fire to be handled after it has got by the door.

34 The spread of fires in the same story is very slow compared with the spread vertically. Fire may be communicated from one story to another by means of well-ways, stairways, or holes in the floors, or by the fire passing out of a window on one story and in at the window above. In a concrete building, the floor forms an ideal fire stop even if the walls have been furred or lathed with wood.

35 There are so many good and cheap metal lathings and furrings on the market today that wood should be prohibited everywhere. While metal does cost more, in the long run it is cheaper to use. With vaulted concrete walls there is no real need of furring or lathing. Concrete construction is particularly well adapted to plastering directly upon the concrete. Furthermore, only one coat is needed to make the same quality of work as two or three coats upon wood or steel lathing. When the plastering is put directly upon the wall, there is no space in which the fire can travel.

36 The regular openings can be protected in the accepted way, and kept free from combustible contents.

37 The spread of a fire to the next building is caused by the combustible gases distilled by the heat from the paints, and varnishes, and even from the wood itself. This explosion (which firemen call hot-air explosion) sets the building on fire in several places at once. While the contents of a concrete building may burn, such a fire will not ordinarily make heat enough to cause the fire to jump wide spaces between buildings. As a further guard the best window, whether translucent or transparent, is one of wired glass. The wires should extend out from the glass sufficiently to be embedded in the concrete sash, holding the glass in place, even when it is heated to a point where a sash of other material would drop the glass or where the glass would fold down and lose its former shape.

38 The disuse of wood in building construction will mean:

a Saving of forests.
b Uninterrupted business.
c Saving of life.
d Saving of buildings.
e Saving of the contents of buildings.

Government Aid in Advancing Fireproof Construction

39 The work to be done to reduce fire losses to a minimum is so great and so important that it can never be thoroughly and completely done until the Government takes it in hand. The Government could aid fireproof construction in various ways.

a By passing laws restricting the use of wood in buildings.
b By levying taxes, discriminating in favor of fireproof houses and against wood in construction.
c By educating the people by Government documents on how to build fireproof houses.
d By establishing a Government bureau for disseminating information regarding honest unbiased fire tests on material, together with Government experiments on different full-size buildings,—kinds, types, materials, etc.—with bulletins of the progress. The cost of this would be but a trifle compared with the benefits. There are several men specially qualified to advise on this work,

as they have been doing such work in their own laboratories for years.

e By building fireproof houses for the use of Government Departments and disseminating information concerning them by means of bulletins.

40 In the meantime, prize competitions would cause special investigation and study in all parts of America for the best house. A few attempts at such competitions have been made in the past. The judges have generally awarded the prizes, in accordance with their best ability, to the designer whose plan pleased them the most. Instead of this, the award should have been made to the competitor who could show the lowest cost per cubic foot, for the finished house with certain specified fireproof requirements. After the standard requirements of a house have been fulfilled, and developed in accordance with the best scheme of fire protection, durability, low cost, permanency speed in building, and comforts in winter and in summer, it is then time to add those beautifying effects that are so necessary from an art standpoint. Such a competition would show to the amazement of many that a better house can we build more cheaply of incombustible materials than of wood.

CONCLUSIONS

41 The only excuse for using wood construction is its low cost. Today—here—now—we have an incombustible material, a material at a less cost than wood that has stood and will stand high temperatures for long periods without injury. Wood must not be used. We do not argue that no other non-combustible material shall be used, and that concrete shall be used exclusively. There are many cases where other non-combustible materials have special merits. They should be used when it is advisable to do so. But now that we have a cheaper and incombustible substitute for wood wood construction, wood trim and wood finish should be legislated and taxed until wood is eliminated from all building construction.

Fig. 1 West Elevation

Fig. 2 East Elevation

Note that the granite was ruined where flames came out at openings. Where the draft was inward, the granite was in perfect condition. The ornamental terra cotta, though apparently but little injured, was completely ruined by heat cracks. Note crack entire height of building, visible in cornice (Fig. 1), showing line where bolts were completely sheared at each story by expansion due to heat.

Cracks extending from top to bottom, about on a line with the clay flue (Fig. 2) mark the location where the bolts sheared. Note also the cracks in the east wall at the fifth-floor level.

FIG. 3 SOUTH ELEVATION

Showing complete destruction of adjoining buildings.

FIG. 4 FALLEN ROOF TRUSS

Structural columns in top story pulled out and broken.

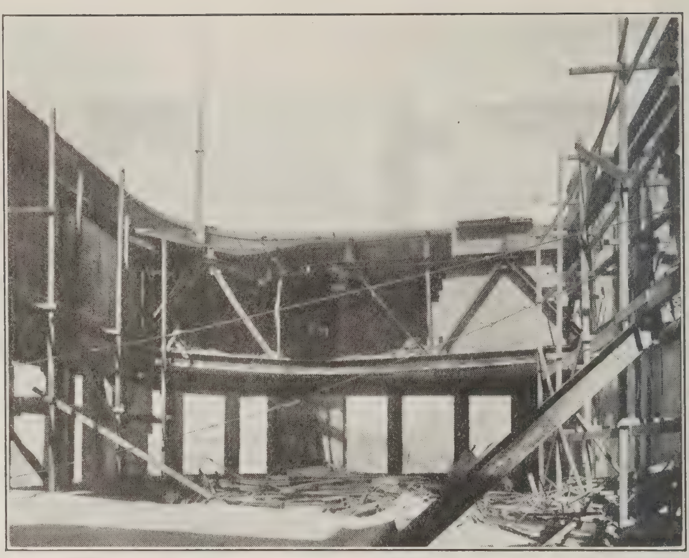

FIG. 5 VIEW LOOKING NORTH IN TOP STORY

Note the complete destruction of the roof trusses and various kinds of material, and that the wooden flagpole was not even scorched.

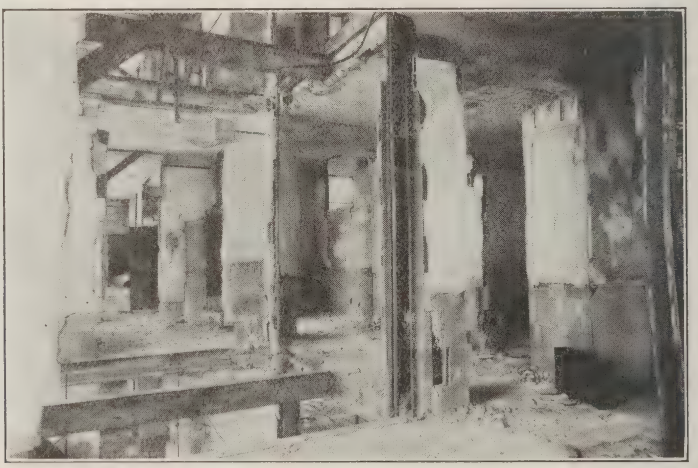

FIG. 6 DAMAGE TO ELEVATOR FRAMING

Although the only combustible material in this building was the wooden trim, floor boards, screeds and furniture, there was sufficient heat to warp the cast iron around the elevators out of shape and to burn up the contents of the steel vaults which were in nearly every room.

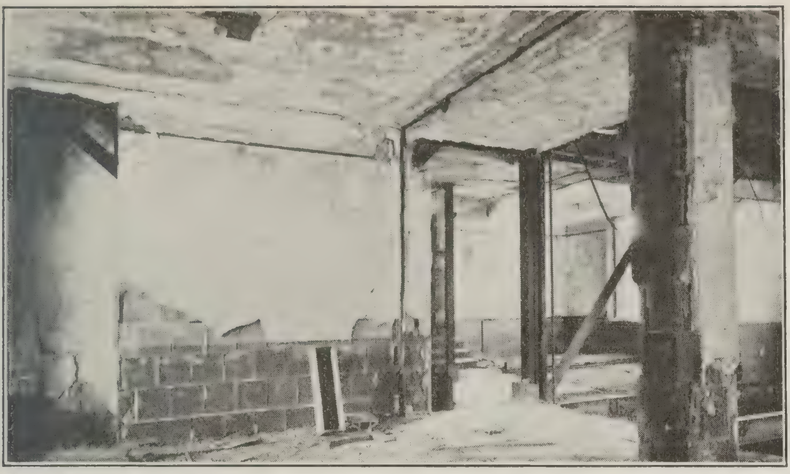

Fig. 7 Destruction of Non-Combustible Material

In all these pictures the columns having pipes buried behind the fireproofing have variably had the fireproofing pushed away from the columns, due to the expansion of the pipes by heat. Note the complete destruction of the marble dado.

Fig. 8 Complete Destruction of Plastering

No kind of plastering has been found to stand the heat of such fire as occurred in this building. Even the plastering that remained upon the wall would not have been strong enough to repair with additional coats.

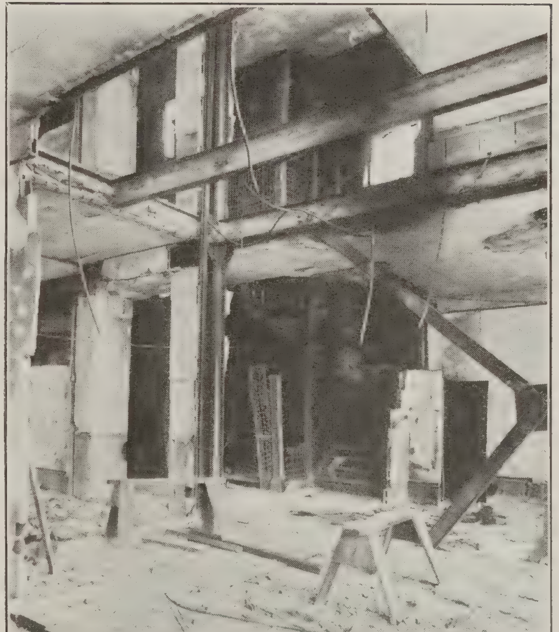

Fig. 9 Large Holes in Floors

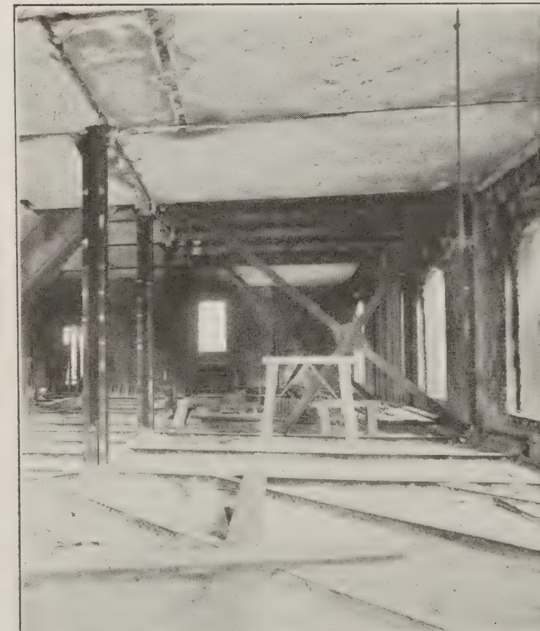

Fig. 10 View of Sixth Story

The large opening between floors (Fig. 9) was caused by structural steel in the roof falling through to the basement.

Fig. 10 is a typical interior view after partitions, cinder concrete fill, and all rubbish had been removed. The bricks in the walls were practically uninjured. The cinder concrete had no strength whatever. The hollow terra cotta floor tile slumped continually, necessitating walking on plank and beams instead of on the floor arches.

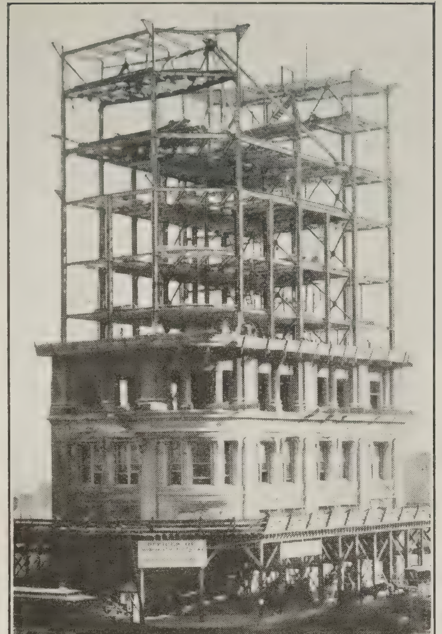

Fig. 11 North Elevation

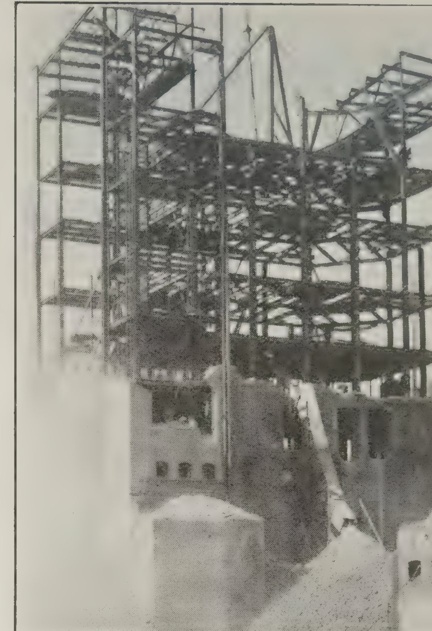

Fig. 12 East Elevation

Steel frame stripped to fourth story. Adjoining two different columns the entire height of the building (Fig. 11), connections had to be rebolted as fast as the work was taken down, asthe bolts were sheared by expansion of the brick walls.

The pile at the left (Fig. 12) shows brick that were sold and used again in new buildings. The rubbish pile of broken bricks, mortar, ornamental and structural terra cotta, cinder filling, etc., was about equal in size to the salvage. The steel below the eighth floor was uninjured. It was sold and used in other buildings in San Francisco.

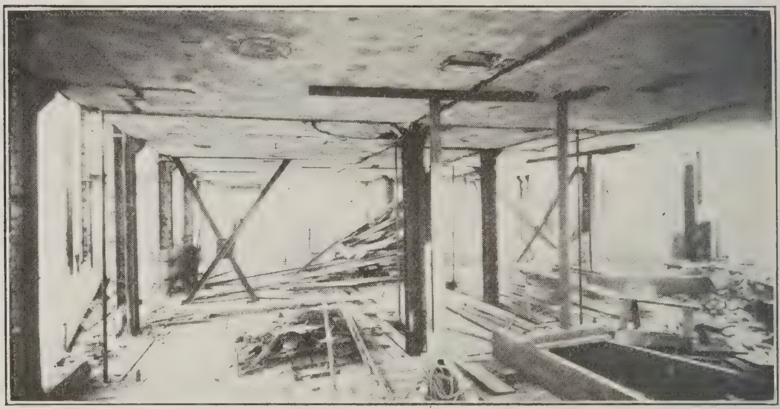

Fig. 13 Destruction of Floors

The small holes in the ceiling were caused by workmen slumping through. The floors were so completely ruined by the heat that it was necessary to cover the third floor with plank to the depth of 18 in., to prevent accidents to the people in the lower stories from the slumping of the floors in the upper stories.

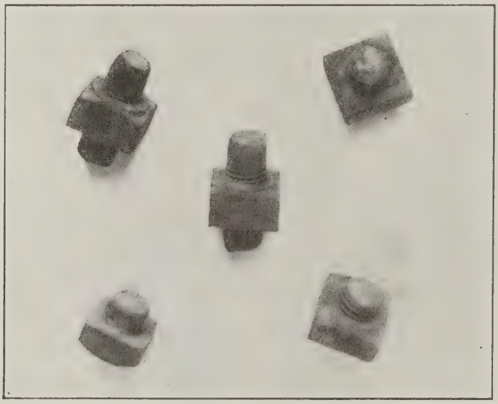

Fig. 14 Samples of Bolts Sheared by Expansion Due to Heat

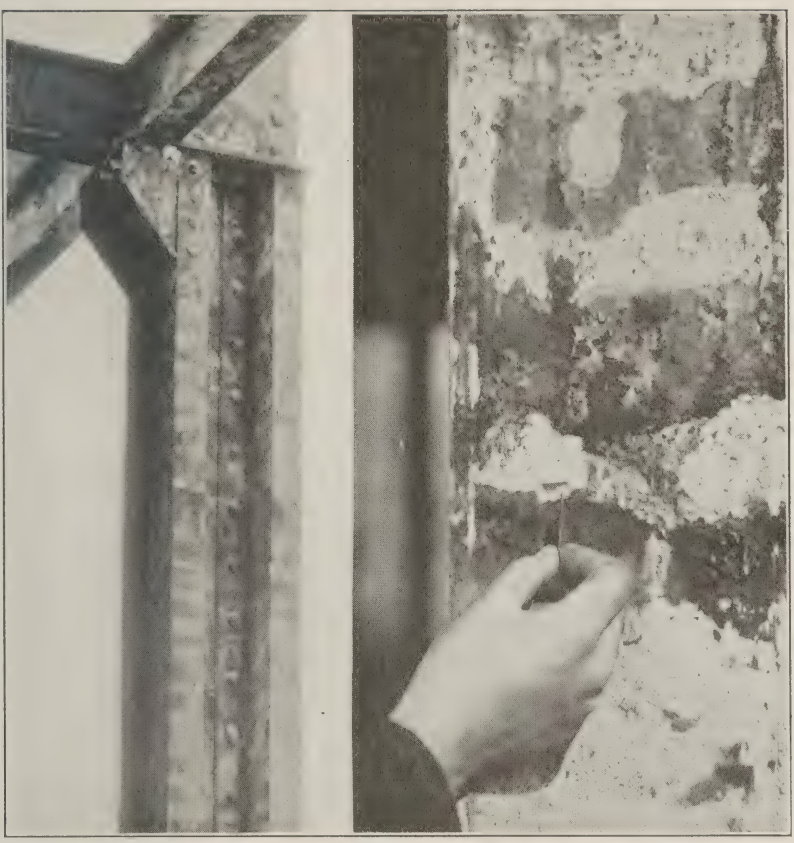

Fig. 15 Condition of Steel Frame

The left side of cut shows a typical exterior floor and column connection. The steel bears no signs of injury or disintegration, although buried in the exterior wall fourteen years. Fire had no effect on the steel, except warping where the frame was not properly protected by fireproofing, but there were several small places, such as that shown on right side of cut, that showed bad rust spots. Each of these cases appeared in steel buried in exterior walls. The rust spots appeared to be places that were not properly cleaned from rust at the time the columns were painted, and were imperfectly surrounded by brick mortar.

THE MECHANICAL ENGINEER AND THE TEXTILE INDUSTRY

By H. L. Gantt, New York
Member of the Society

The textile industry enjoys the distinction, to a greater extent, perhaps, than any other, of having been brought to a high state of perfection without the aid of the mechanical engineer. The machinery was developed by the mechanic before the mechanical engineer became a very important factor in the industrial world, and the plans were, and are still, built by mill architects, who as their name implies are architects rather than engineers. The most important field of this industry that the engineer has entered, is the power department, and in this field he has done much good work. The complicated and delicate machines for working cotton fibre, however, which are wonders of mechanical skill, have been brought to their high state of perfection by men who were mechanics rather than mechanical engineers. The operation of these machines, until recently, has been directed by men whose training was exclusively that of the factory, and who could solve well a concrete problem.

2 In this industry there is, as a rule, a wider gap between the financial interests that control, and the "help" that operate, than there is in almost any other industry.

3 The textile schools are today doing much to fill this gap by supplying to the mills educated men who, while understanding the detail operation of the machines, are capable of comprehending the larger problems of management, and can thus form a link between the financial men that control and the mechanics that operate.

4 The lack of such men in the past is undoubtedly responsible for the fact that some of the processes which influence the subject of management more than they do the product, and which are easily susceptible of being standardized and done automatically, are

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still being done expensively and inefficiently by hand, and in a manner that causes much subsequent labor and expense that should be avoided. The solution of this problem belongs more particularly to the function of management, for the workman often does not see the influence of one process on a subsequent one in another department.

5 I refer, for example, to the process of handling cloth in a bleachery. In order to make clear the point in mind, it is necessary to explain that in the process of bleaching, cotton cloth is generally sewed together piece by piece and handled in the form of a rope, which is drawn from one operation to the next by means of rolls. This rope of cloth is subjected to the action of various liquids, being first boiled in an alkali and then washed. After being washed it is usually impregnated with acid (technically "soured"), and allowed to stand in a pile for some minutes to allow the acid to act. The methods of forming this pile and of withdrawing the cloth from the pile, are the operations to which I have special reference.

6 As the piling operation is repeated after each of several impregnating operations, the successive pilings divide the process into a series of separate and distinct stages with a loss of time between every two. The usual method of piling is as follows:

7 The cloth is drawn from the souring machine by an overhead roll, which drops it to the floor beneath. A boy stands on the pile of cloth and so guides it with a stick that it is piled in substantially uniform horizontal layers. When the pile has reached a size determined by the judgment of the bleacher (or the boy), the rope of cloth is broken at a seam and a second pile is formed. When in the judgment of the bleacher the first pile has stood long enough, the cloth is withdrawn and pulled through a washing machine into a bath of chlorine water (technically "chemic"), after which it is again piled in the same manner by a boy with a stick. The judgment of the bleacher as to the time cloth should lie in a pile after impregnation seems to be controlled by his temperament, or by tradition, rather than by knowledge, for we find that hardly any two bleachers have the same opinion as to how long the cloth should be subjected to the action of the acid; and the practice varies from a few minutes to twenty-four hours. As a matter of fact the acid does all its work in ten minutes or less, and no beneficial effect can be discovered by a longer treatment.

8 Inasmuch as it is necessary to pull the cloth from the top of a pile, the leading portion as it leaves the sour pile has been acted

upon by the acid a shorter time than that at the bottom of the pile.

9 The top of the second pile is attached to the bottom strand of the first pile, and the top of the third pile is attached to the bottom of the second.

10 As each strand of cloth usually goes through several pilings in the course of being bleached, the action of the bleaching liquors on any portion of the cloth would be alternately long and short, according as that portion of the cloth was at the bottom or the top of a pile. If the rope of cloth was always broken in the same place, the worst that could happen would be an unevenness in the bleach due to the difference in treatment. It frequently happens, however (and this is more often the case than not), that the rope of cloth is not broken in the same place; and when this occurs the various lots of cloth of which the rope is composed, which usually belong to different customers become almost hopelessly mixed. The expense of straightening out such a mix-up has usually been considered one of the legitimate expenses of bleaching. Add to this the fact that the piling boy often piles the cloth so carelessly that it tangles as it is pulled off the pile, and not only damages itself, but usually shuts down a portion of the plant for awhile.

11 If we also realize the fact that chlorine, or "chemic," not only forms a most unpleasant atmosphere to work in, but is actually injurious to the lungs, it would seem that some automatic piling machine which would hold the required amount of cloth and permit the leading end of the pile to be withdrawn would long ago have been devised. Inasmuch, however, as this is not a problem requiring great mechanical skill, but one requiring a somewhat different kind of knowledge, it apparently had never been attacked until the writer came in contact with it.

12 Fig. 1 shows the machine which has been developed to accomplish the result, and Fig. 2 shows the cloth as it is delivered to and withdrawn from the machine. The machine consists of an inclined chute, with upturned ends, and having a bottom composed of a series of independent rollers, freely revolving. The cloth is dropped into the tall stack, and falling on the rollers is carried by its own weight to the bottom of the incline. The incline is filled, and as the fabric rises in the receiving stack, the forward end of the pile is forced upward in the other end of the machine, from which it is pulled off at the rate at which it enters the receiving stack.

13 By making the chute of the proper length a pile of cloth of any size may be held, and the cloth may be subjected to the action of the

impregnating liquor for any desired time, all portions of the fabric receiving exactly the same treatment. Such action produces uni-

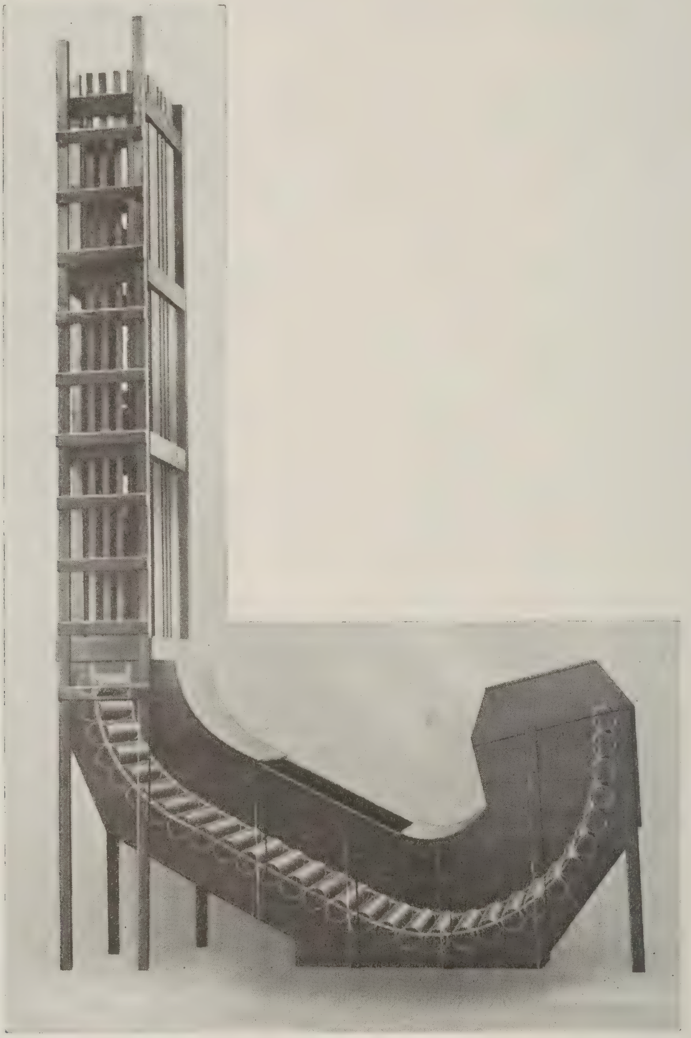

Fig. 1 Automatic Piling Machine

formity of bleach impossible under the old conditions, and as there is no need for breaking seams, the goods go through the bleach house in the order they went in, which produces a saving of expense and

worry realized only by the man who has operated under both methods. The straightening out of "mix-ups" and the "closing out" of "short lots" are the bane of a finisher's existence, and anything that reduces these troubles does much, not only to smooth the operation of the works, but to assure the customer that he is

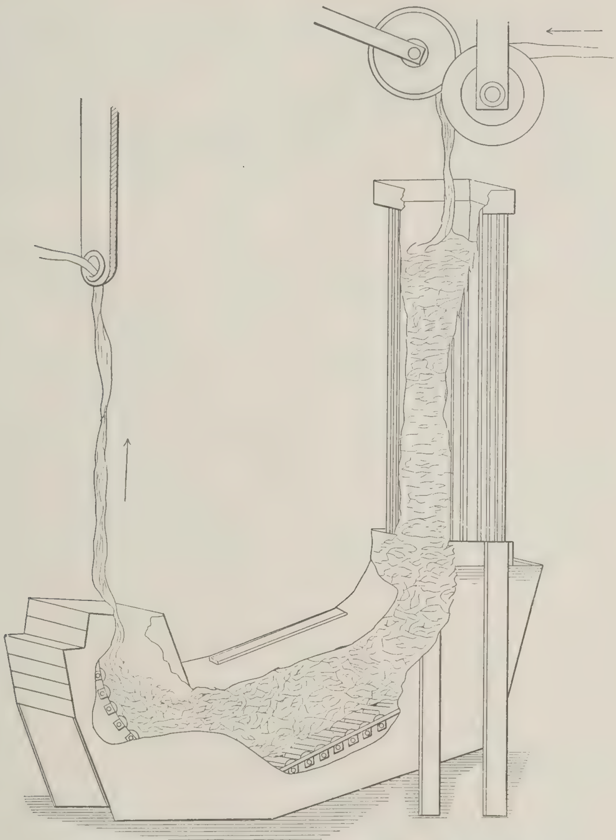

Fig. 2 Progress of Cloth through Machine.

getting back exactly the goods he sent. Moreover the dirt and damage caused by the piling boys are eliminated.

14 The saving in always having clean goods in uniform condition is greater than the saving in wages of the boys, and the relief to the foreman of having a smaller number of bleach-house boys to manage, makes it possible for him to devote his time to bleaching rather than to boys, with distinctly beneficial results to the bleaching.

15 In addition to the advantages already mentioned, there is a marked saving in time, for the cloth remains subject to the action of each liquid only the time needed to produce the desired result. Each piling machine takes the place of from three to four bins, and as it takes up less space than one bin, the saving in buildings is very con-

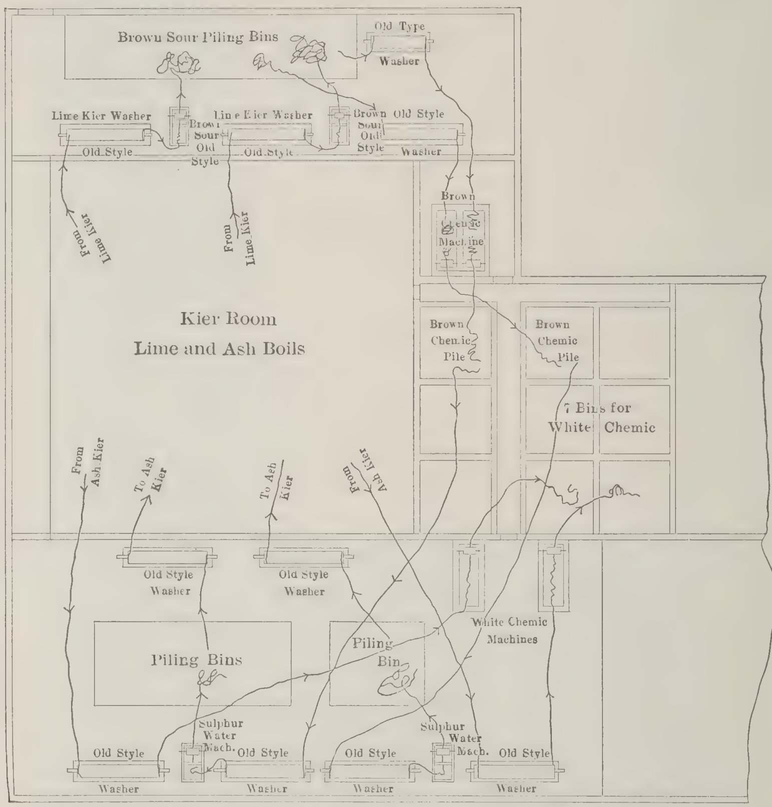

FIG. 3 ARRANGEMENT OF BLEACHERY (OLD SYSTEM).

siderable. In one case, when the washing, "souring" and "chemic" machines were rearranged in such a manner as to use a full equipment of piling machines to the best advantage, the saving in bleach-house space amounted to more than 40 per cent. Wherever the machines have once been installed it is obvious that they soon be come indispensable.

16 The fact that such an important operation can be taken care of in such a simple manner, is the best evidence that the writer entered

a field that has not been thoroughly investigated by the mechanical engineer. The field is still open, for plants are being built today to handle cloth exactly as it has been handled for fifty years. The builders of these plants have not yet discovered the function of the mechanical engineer, and are still putting their faith exclusively in bleachers.

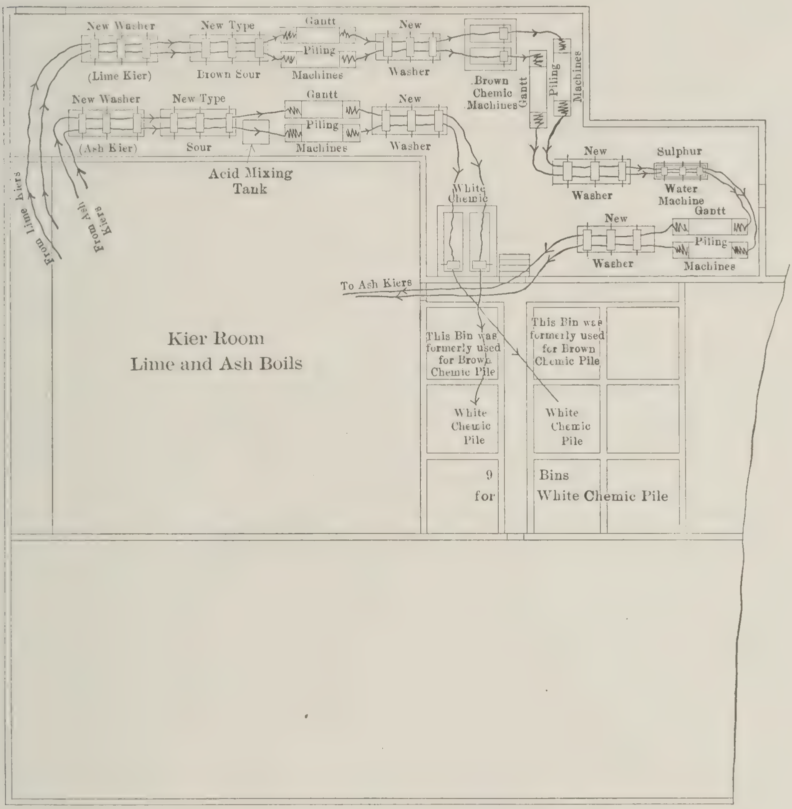

Fig. 4 Arrangement of Bleachery (New System).

17 Fig. 3 represents the course of cloth through a bleachery where the writer was told that the process had not been changed for fifty years. Fig. 4 shows the course of the cloth in the same bleachery after it had been equipped with piling machines, and other machines adapted to work economically with them. The new installation takes up less than 60 per cent of the space of the old and is operated by six people against the twenty people formerly needed.

18 The operations and their sequence are the same in the new as in the old lay-out, except for the installation of an additional "sour", which was thought desirable to remove the alkali of the second boil. The new lay-out is not a new method of bleaching, but simply a mechanical engineer's method of handling the old. There is hardly a bleachery in the country where the mechanical engineer cannot do similar work, and that without doing violence to the prejudices of the bleacher.

19 The standardization of bleaching methods must come later, and will take time, for we have here the habits of at least half a century to combat.

BALL-BEARING LINESHAFT HANGERS

By Henry Hess, Philadelphia, Pa.
Member of the Society

This paper is presented because of a request for more specific information made during the discussion of a paper that I was privileged to read at the Annual Meeting in New York, December 1909. That paper dealt with Lineshaft Efficiency, Mechanical and Economic, and was restricted to the presentation of actual measurements of the power consumption of the same lineshaft when mounted on plain and again on ball bearings, with that change the only variable.

2 It was the purpose of the paper to describe merely the actual test and its results, thus adding to the fund of available engineering knowledge. The favorable economic showing of the ball-bearing is inherent in the advantages of rolling as compared with sliding friction, quite aside from any particular make of ball-bearing; though if durability as well is to be secured, it naturally is essential that bearings of correct design and suitable workmanship and material be selected. The demand at the time of the discussion and since must serve as apology for such references to a product for which I am responsible as are incident to a description of the installation in question.

3 The first real improvement in hangers was made in this country by one of the oldest and most honored members of our Society. It was Mr. Bancroft who first mounted the box to swivel fully while well supported and who also gave it vertical adjustability. The means are so familiar to every one today as to constitute a commonplace. This original hanger was supplied with carefully fitted adjusting screws of liberal size; their ends terminated in correctly machined spherical sockets fitting over correspondingly machined spherical segments on the top and bottom of the box. Once adjusted in the hanger the box retained its place until a readjustment was demanded by some outside cause such as a settling building.

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4 Unfortunately the constant and incessant demand for lower and yet lower costs was given heed to by many until in course of time not a few of the many makes of hangers marketed are not machined at all but are foundry jobs throughout. Even so the consequent looseness and indefiniteness of position of the box supporting screws is not a very serious matter with plain bearings. The very length of the plain

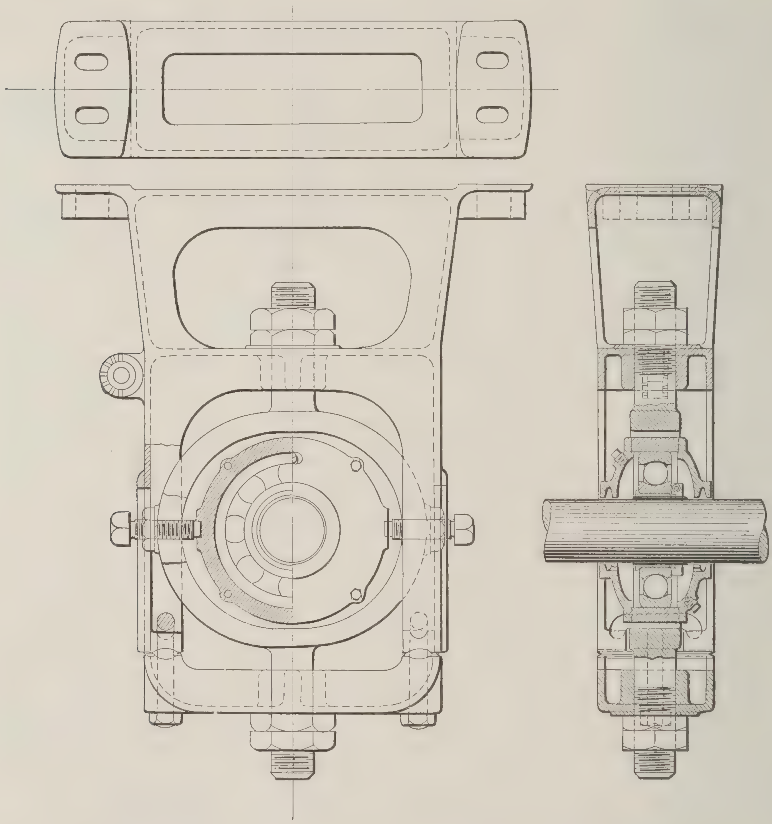

Fig. 1 Arrangement of Ball-Bearing Hanger

bearing equal to some four or five shaft diameters helps to hold the box in line with the shaft if that is straight; or if the deflecting forces are too great, then the resultant excessive pressure on the plain bearings will draw attention to the trouble and by an insistent squeal, demand correction.

5 In first applying ball bearings to lineshafts I employed variou

hangers such as were on the market. The ball bearing was set in an enclosing box which in turn was held between the same supporting screws that supported the plain bearing displaced. Occasional reports were received that the power consumption was greater after the change than before. These were particularly insistent from a few installations where accurate meter records of electrical power consumed by the shaft driving motors were kept. Investigation finally disclosed in each instance a shifting of the shaft sections from their original alignment. The consequent and in several cases quite severe binding of the shaft at each revolution consumed considerable power. Owing to the very low coefficient of ball bearing friction, 0.0015, however, the additional load on the ball bearings was without evident effect and caused neither squealing nor even perceptible heating. Realigning the boxes immediately brought down the power consumption to the original low figure.

6 The Bancroft, or as they are better known, "Sellers" type hangers with their definitely machined supporting screws and sections of spheres on the outside of the box to rock on, was not found practicable with ball bearings, because these spherical sections became too flat with the large diameter of the box containing the ball bearing, as is evident from an inspection of the cross section of such a box given in Fig. 1.

7 It was finally and reluctantly decided to make up and build hangers, general coöperation with the specialists in that line having been found impossible from lack of interest, reluctance to make changes, the usual inertia resisting innovations, etc.

8 Many designs were laid out on paper and some tried, but again discarded as not responding to all of the following requirements of an ideal hanger:

a Definite support of the box on machined seats permitting no shifting under load.

b Ability of the box to swivel in all directions.

c Vertical adjustability of the box within the hanger body.

d Horizontal adjustability of the box within the hanger body.

e Rigidity in every direction, to permit the hanger to be used as a ceiling hanger, floor stand or post hanger.

f Convenience of adjustment in aligning the shaft.

g Adaptability for countershaft hangers involving a shifter-arm.

h Neatness of general outline and conformity to modern machine design by substituting box sections for ribbing.

i The lowest cost consistent with the other requirements.

9 The design finally evolved is shown in Fig. 1. The box is supported and pivoted horizontally on the ends of two screws tapped through the opposing sides of a yoke surrounding the box with enough freedom to give 1 in. of horizontal adjustment. This yoke has an upper and lower cylindrical stem, each turned to a running fit in the corresponding bore of the hanger; these stems are threaded and provided with nuts and checknuts and allow a vertical adjustment of 2 in. As the box can swivel horizontally in the yoke and that in turn vertically in the hanger the two together provide the desired universal swivelling freedom.

10 The hanger body is of channel section with the flat outside. The central crossbar that takes the weight of the box and shaft also materially stiffens the hanger. The lower crossbar is cast integrally with the hanger body and split off in the usual way. A single bolt at each end serves for attachment. This bolt is a drop forged T with the T turned down and ball ended to fit in corresponding sockets.

11 All of the various sizes, with the exception of the largest, which never comes into consideration for countershafts, are provided with cast-on lug with a serrated side face. To this may be bolted an arm of suitable length at any desired angle, to carry the usual eye guiding the shifter rod.

12 The construction of the bearing box is also apparent from Fig. 1, which shows it to consist of a central cast supporting ring bored to a sucking fit for the outer race of the ball bearing. To the side faces coverplates are bolted with an oil-tight joint and where these surround the shaft they are provided with a cored annular groove. The side lips are bored $\frac{1}{64}$ in. larger than the shaft diameter and have sharp edges.

13 This arrangement is found to be efficient in retaining the lubricant and so promotes cleanliness, while also preventing the entrance of ordinary foreign matter. For particularly difficult locations, as in cement mills, the very fine floating grit is kept out by a double groove, the outer one of which is filled with a fairly heavy grease. A single charge of heavy cylinder oil or non-acid grease will last the average lineshaft bearing several years; and to avoid total neglect refilling once a year is recommended.

14 The ball bearings are free endwise in the box. It is recommended that a lineshaft be held endwise by ordinary collars on either side of the central hanger so that all weaving endwise, due to expansion and contraction, settling, etc., is allowed for by the end freedom in the boxes. An occasional installation is found with heavy end-thrust, in which case the thrust is taken on one of the ball bearings by letting

it come up against the shoulders of the coverplates. Unusually heavy thrusts are provided for by special hangers into which a collar type of ball bearing is built.

15 As it is necessary to clamp the ball bearing to the shaft so that the inner race cannot rotate and as commercial shafting is somewhat indeterminate in size, a so-called "adapter" is employed, the details of which are shown in Fig. 2. The adapter consists of a bush fitting into the bore of the inner race, which in turn is fitted with a coned split sleeve. This latter is driven home endwise until the whole is tightly

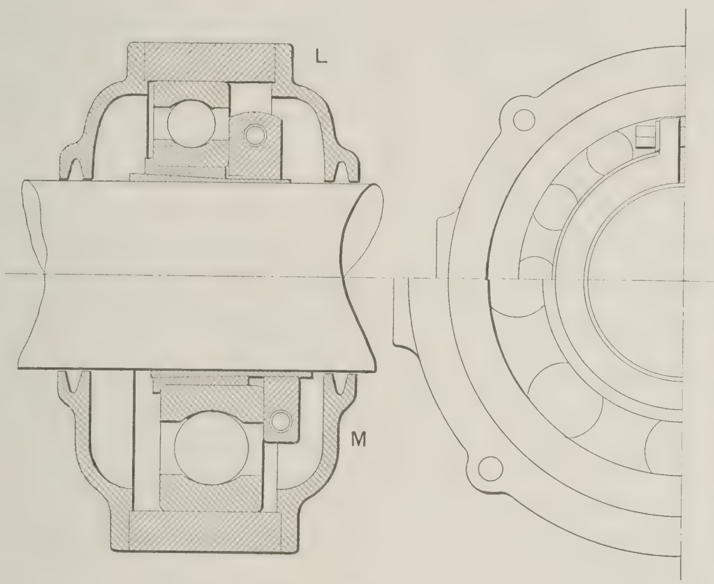

Fig. 2 Ball-Bearing Adapter

clamped onto the shaft retaining a truly concentric setting. Unless there is considerable vibration, the wedging effect of the small angle of the conical bush is sufficient to prevent any loosening, but to guard against all contingencies a split collar is clamped onto the sleeve with its face close up to the face of the bearing.

16 In the sectional view, Fig. 2, that part of the bearing above the center line appears narrower than that below, the first marked "L" and the latter "M." This indicates the relative widths of light-weight and heavy-weight bearings. Bearings of the "L" or light-

weight series take less load than those of the "M," or medium-weight, series. For all ordinary lineshafts the light-weight series is quite sufficient; it is only for extraordinary conditions or for jackshafts, etc., that the medium series comes into consideration. A single medium bearing is sometimes also used in the hanger next to the main belt.

17 The ball bearing itself is a very simple element, consisting, as is clear from both illustrations, of an inner race, an outer race, and a single row of interposed balls, running in grooves whose radius of curvature is only a few percent larger than the ball radius. Races and balls are made of alloy steels (a special mixture of the carbon, chrome, manganese variety).

18 These ball bearings are the same that are regularly supplied for the machine industries in general. Adapters for shafting range from $\frac{15}{16}$ in. to 3 $\frac{15}{16}$ in., both light and medium. As larger shafts are rarely of the nature of lineshafts, it is preferable to fit seats directly to the bearing bore without the interposition of adapters.

19 In the paper referred to at the outset direct measurements were cited showing a saving in lineshaft friction of about 35 per cent under proper belt tensions of 44 lb. to 57 lb. per inch width of single belt and correspondingly more for heavier loads. The return on the investment was also shown to be 37 per cent per annum, taking into account the higher cost of the ball-bearing installation. Such higher cost is, however, not necessary. The initial cost may, in fact, be actually lower if in the original layout full advantage is taken of the possibilities of the ball bearing. The ball bearing is as safe and reliable at 600 r.p.m. as at 200; whereas, the use of such high speed with plain bearings is beset with so many difficulties in the way of reduction of size of lineshaft, pulleys, drop of hangers, etc., as to take that practically out of consideration for the average plant. The first cost of a ball bearing installation of 600 r.p.m. compares favorably with the first cost of one at 200 r.p.m. on plain bearings and the full advantage of the saving to be derived from ball bearings may thus be realized without any extra investment whatsoever.

20 In fact, the first cost of the ball-bearing equipment is quite likely to be less. As an example, it is fair to take a lineshaft such as that on which the test referred to in the December paper was made, but fully loaded to drive 32 machines. The installation elements that vary with the shaft speed are the shaft diameters, pulley diameters, hanger drops, belt widths and countershaft-driven pulleys.

The net cost of the installation material for the slow speed shaft of 200 r.p.m. on plain bearings transmitting 40 h.p. is	$558.95
The net cost of the alternate installation for the high speed shaft of 600 r.p.m. on ball bearings is	441.87
The saving in first cost of material amounting to 21 per cent	$117.08

See Appendix for details.

21 The lineshaft bearing friction saving, based on an average coefficient of friction and confirmed by the series of tests cited in the previous paper, is 0.41 kw. or 1230 kw.-hr. for a year of 3000 working hours and worth, at 3 cents per kw.-hr., $36.90; secured by an installation 21 per cent lower in initial cost. See the Appendix for further details.

APPENDIX

COST OF PLAIN-BEARING SLOW-SPEED LINESHAFT MATERIAL

Shaft 200 r.p.m. in ring-oiling boxes, transmitting 40 h.p., driving 32 countershafts, one belt to each countershaft, 8 ft. to countershaft centers, countershafts 300 r.p.m.

10 hangers, 16-in. drop, ring-oiling babbitted boxes, net cost..........			$53.63
1 main pulley, 40 -in. x $12\frac{1}{2}$-in. double belt			
4 line pulleys 15 -in. x $6\frac{3}{4}$-in. single belt	4 counterpulleys	10 -in. x $6\frac{3}{4}$-in.	
4 " " $8\frac{3}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	6 -in. x $4\frac{1}{2}$-in.	
4 " " 10 -in. x $4\frac{3}{4}$-in. double belt	4 counterpulleys	$6\frac{3}{4}$-in. x $4\frac{3}{4}$-in.	
4 " " $9\frac{7}{8}$-in. x $5\frac{1}{2}$-in. single belt	4 counterpulleys	$6\frac{1}{2}$-in. x $5\frac{1}{2}$-in.	
4 " " $9\frac{3}{4}$-in. x $7\frac{3}{4}$-in. double belt	4 counterpulleys	$6\frac{1}{2}$-in. x $7\frac{3}{4}$-in.	
4 " " $8\frac{3}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	6 -in. x $4\frac{1}{2}$-in.	
4 " " 10 -in. x $4\frac{3}{4}$-in. double belt	4 counterpulleys	7 -in. x $4\frac{3}{4}$-in.	
4 " " $12\frac{1}{4}$-in. x $4\frac{1}{2}$-in. single belt	4 counterpulleys	8 -in. x $4\frac{1}{2}$-in.	
	Net cost................		117.97
20-ft. 12-in. double belt			
320-ft. 3-in. single belt			
160-ft. 3-in. double belt			
80-ft. $3\frac{3}{4}$-in. single belt			
80-ft. 4-in. double belt			
	Net cost................		360.10
72-ft. $2\frac{7}{16}$-in. lineshaft	Net cost................		27.25
		Total net cost.....	$558.95

COST OF BALL BEARING HIGH SPEED LINESHAFT MATERIAL

Shaft 200 r.p.m. in Hess-Bright medium series ball bearings, transmitting 40 h.p., driving 32 countershafts, one belt to each countershaft, 8 ft. to countershaft centers, countershafts 300 r.p.m. Lineshaft pulleys reduced in diameter by one-half and the driven countershaft pulleys incerased in diameter by one-third; belts decreased in width in correspondence with the increased speed; lineshaft decreased in diameter proportionately to speed; under both conditions safe for 60 h.p. according to catalog ratings. Hanger decreased in drop to 10 in., rather more than the diameter of largest counterdriving pulley. Belts 50-lb. per inch width, single.

10 hangers, 10-in. drop, medium weight ball bearings, net cost........			$159.20
1 main pulley, 28-in. x $6\frac{1}{2}$-in. double belt			
4 line pulleys $7\frac{1}{2}$-in. x $4\frac{1}{2}$-in. double belt	4 counterpulleys 15-in. x $4\frac{1}{2}$-in.		
4 line pulleys 6-in. x $3\frac{1}{2}$-in. single belt	4 counterpulleys 12-in. x $3\frac{1}{2}$-in.		
4 line pulleys 5-in. x $2\frac{1}{2}$-in. single belt	4 counterpulleys 10-in. x $2\frac{1}{2}$-in.		
4 line pulleys 5-in. x $2\frac{3}{4}$-in. single belt	4 counterpulleys 10-in. x $2\frac{3}{4}$-in.		
4 line pulleys 5-in. x $4\frac{1}{2}$-in. double belt	4 counterpulleys 10-in. x $4\frac{1}{2}$-in.		
4 line pulleys 6-in. x 2 -in. single belt	4 counterpulleys 12-in. x 2 -in.		
4 line pulleys 5-in. x 3 -in. single belt	4 counterpulleys 10-in. x 3 -in.		
4 line pulleys 8-in. x 2 -in. single belt	4 counterpulleys 16-in. x 2 -in.		
	Net cost................		87.57
20-ft. 6 -in. double belt			
240-ft. 2 -in. single belt			
240-ft. $1\frac{1}{2}$-in. single belt			
80-ft. $2\frac{1}{2}$-in. single belt			
80-ft. $2\frac{3}{4}$-in. single belt	Net cost................		162.00
72-ft. $1\frac{11}{16}$-in. lineshaft	Net cost................		13.10
	Total net cost...........		$441.87

ESTIMATE OF SAVING IN KILOWATTS

In the Author's paper, on Lineshaft Efficiency, published in The Journal for December 1909, Par. 32, is given: Kw = $0.0000059\ Lds\mu$

		Plain Bearings	Ball Bearings
L	= total journal load in lb. =	9606	5420
d	= shaft diameter in inches =	$2\frac{7}{16}$-in.	$1\frac{11}{16}$-in.
s	= shaft speed in r.p.m. =	200	600
μ	= coefficient of friction =	0.03	0.0015
Kw	= kilowatt friction loss =	0.90	0.49

Kw., plain = $0.0000059 \times 9606 \times 2\frac{7}{16} \times 200 \times 0.03 = 0.90$

Kw., ball bearing = $0.0000059 \times 5420 \times 1\frac{11}{16} \times 600 \times 0.0015 = 0.49$

Kw. saving = $0.90 - 0.49 = 0.41$

THE HYDROSTATIC CHORD

WITH DISCUSSION OF ITS APPLICATION IN THE DESIGN OF LARGE PIPES OF REINFORCED CONCRETE

By Raymond D. Johnson,[1] Niagara Falls, N. Y.

Non-Member

The hydrostatic chord is allied to the catenary, the parabola and the circle, because all of these curves may be formed by a flexible inextensible substance, supported at its two extremities and properly loaded. If the load is uniformly distributed with respect to a horizontal line joining the supports, the action of gravity will shape the supporting substance to form a parabola. If the load is uniformly distributed with respect to the curve itself a catenary is the result. If the load is applied by fluid pressure, irrespective of the direction of gravity, so that the pressure is of uniform intensity normal to the curve, a circle is formed. If the load is applied by fluid pressure which varies according to the head or depth of water at any point, the curve resulting from this system of forces normal to the curve is a hydrostatic chord, which can easily be imagined as the curve which a flat canvas hammock would take if filled with water.

2 If the canvas were sewed together to form a closed curve, and supported on end as a vertical cylinder, the cross section would become circular under fluid pressure. If now the open ends of the cylinder were sealed with flexible bulkheads and the cylinder was tipped over on its side when completely filled with water the cross section would become a hydrostatic chord, although it would still theoretically be a circle also, until a drop of water was allowed to escape, that is, if the water be regarded as incompressible. Since the shell of the cylinder is assumed inextensible, and a circle encloses the maximum area for a given perimeter, it follows that the mere act of tipping such a cylinder towards the horizontal position would immediately develop infinite stress in the enclosing membrane, if the water could not par-

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[1] Ontario Power Company, Niagara Falls, N. Y.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

tially escape. If one now imagines a hole pricked in the side of the cylinder on top, and connected with a vertical pipe or piezometer, some of the water would escape up into the pipe and at the same time the shape of the cylinder would take on the characteristics of a true hydrostatic chord. The water would continue to rise until the head above the top was just sufficient to hold the remaining water in the equilibrium shape. At this time the surrounding membrane would be in pure tension, equal throughout, and of finite value. The very fact that the tension must be constant in all portions of the shell furnishes an easy means of constructing its shape, because the tension at any point is obviously measured by the product of the radius or curvature and the head at that point.

3 If more water now be poured into the piezometer pipe, the shape of the cylinder will approach a circle; its vertical diameter will lengthen and its horizontal diameter will shorten. It would, however, never become a circle for a finite head of water. Conversely, if water be drawn off, the shape would become more and more oblate until finally the membrane would collapse into a plane surface at zero head.

4 It seems apparent from the above discussion that a circle is not the natural shape for a pressure pipe lying on its side, especially if its diameter is large as compared with the water pressure. No one would think of designing a vertical water tank with an elliptical cross section, and thus subjecting the shell to enormous and unnecessary deforming stresses in its effort to become circular. And yet it has not been unusual, not only to design concrete pressure pipes circular, but even to go to the other extreme and shape the section with its least radius of curvature at the top instead of at the haunches.

5 This latter procedure might be compared to designing a stiff suspension bridge cable, say, for the sake of illustration, of reinforced concrete, and shaping it like an ellipse with its long axis horizontal, instead of the more natural shape in which the radius of curvature would decrease toward the center, instead of increasing. Such a chord is obviously so ridiculous that an example of it could not be found in practice. Instead it would probably be designed of parabolic shape, which would be perhaps as good a compromise as one could reach. If it were really too stiff to adjust its shape to changes of load, as for example when a moving load passed over the bridge, then more or less severe deforming stresses would be the result, but how much less than in the former case.

6 Similarly, although it is impossible to design a stiff pipe of

such a shape that there will be no deforming stresses under the varying conditions of water pressure and back fill and the constant weight of the shell itself, yet these stresses can be reduced to a minimum by adopting a form which lies midway between the ideal equilibrium shapes which the pipe tries to assume under the various water pressures to which it may be subjected. This matter is of much practical importance in large conduits, proper design of the cross section meaning the saving of perhaps one-half the material, concrete or steel.

7 It is not the writer's purpose to take up the mathematics of the hydrostatic chord, nor to follow through the complications of a typical design. The element of judgment enters so largely into such a study that it is impossible to do it justice in restricted space and time. A few general hints may be of service.

8 It is well known that a circular cylindrical shell lying on its side has four nodes, or points of contra-flexure, due to its own weight. These lie at points 50 deg. 36 min. 45 sec., and 146 deg. 19 min. 25 sec., respectively, from the vertical. It can be demonstrated that the locations of the nodes due to the weight of the water within such a cylinder are the same. It can also be shown that the bending moments due to both causes are exactly proportional at all points of the arc, and may therefore easily be combined. The equilibrium shape which would sustain the existing water pressure without any tendency to deform, can easily be plotted from the polar equation of the bending moments in a circle, in terms of the angle of departure from the vertical, remembering that the radial intercept between the circle and the new curve at any point, is a measure of the bending moment at that point, and when multiplied by the corresponding tension at that point of the circle will give the value of the bending moment.

9 Conversely, if the bending moment be divided by the tension, the radial intercept will be the quotient, and may be plotted. The value of this intercept at any angle, and for any assumed head H above the top of the pipe, is as follows:

$$\frac{r\left(\frac{1}{2} - y\right)}{\dfrac{H + r}{r} - y}$$

where r = radius of the circle, and $y = \frac{1}{4} \cos \phi + \frac{1}{2} \phi \sin \phi$.

10 Any number of such curves may be plotted, according to the number of different values of H assumed, and all of these will of course pass through the same node points.

11 These equilibrium shapes are not identical with the hydrostatic chord, for the reason that the forces acting to produce the latter are strictly normal to the curve itself, whereas in the former case the applied forces are always considered to be normal to the common circle for all the different equilibrium curves.

12 The discrepancy between the two curves for any given head, is, however, so slight as to be negligible for all practical purposes. A measurable difference would scarcely be found except for very low heads, say less than one diameter above the top of the pipe, where the stresses are small and not important. After the head reaches

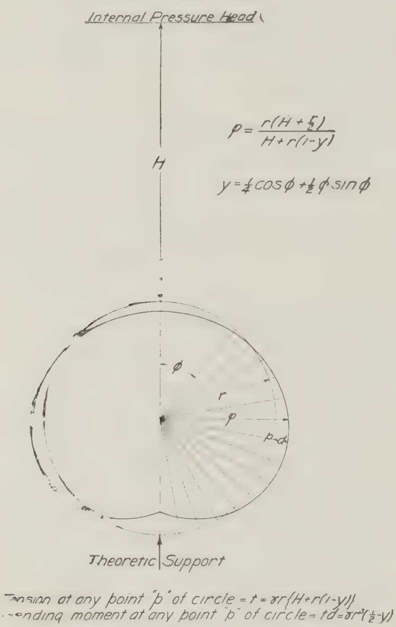

Fig. 1. Equilibrium Curve

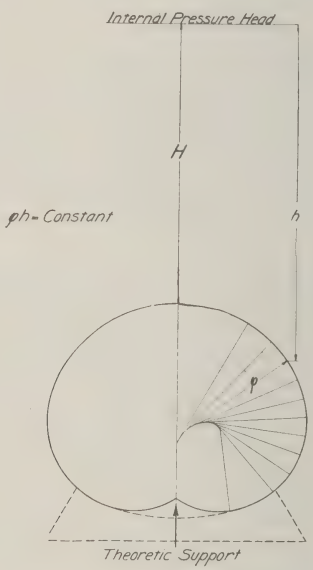

Fig. 2 Hydrostatic Chord for Same Head

a value of five or six diameters above the top of the pipe, the departure from a circular shape is comparatively slight, and therefore this discussion is particularly applicable only to large pipes under low pressure.

13 It will be found that when the pipe is supported on a continuous saddle, as is usually the case, the maximum stress is likely to be located at the top of the pipe, so that if the shell is made homogeneous no other point need be investigated for stress. If the pipe, in-

stead of being circular, is formed on the lines of any one of the equilibrium curves, or better still, on the lines of the true hydrostatic chord for a particular head, then the bending moments induced in it by any other head may be scaled or computated at any point by noting the length of the intercept between such a curve and the curve corresponding to the head under consideration. The value of the tension at any point, which must be multiplied by this intercept, is given by the formula $\alpha r\,[H + r\,(1 - y]$, where α is the weight of a cubic foot of water and the other quantities remain as before.

14 In arriving at the total stress it is necessary to combine algebraically the bending stresses due to weight of shell, back fill and weight of water, and then add to them the tensile stress due to the water pressure. The bending moments in a circle when lying on a flat surface are simply expressed by the equation, $-\alpha\, r^3\,(\frac{1}{2} - y)$.

15 When the pipe rests on a saddle the maximum stress is usually found at the top of the pipe, as before stated; and although its amount is somewhat lessened by the presence of the saddle underneath, it is not considered advisable to rely on this, and it is best to design for stress at this point strictly as though the pipe had a very narrow saddle, or theoretically none at all.

16 The above reasoning is not to be regarded as hard and fast for text-book use, and the conclusions are known to be merely close approximations. Many interesting properties of the hydrostatic chord have been studied by the writer, and much of the mechanics relating to pipe design has been more or less thoroughly worked out; but there is so much of no practical application, and so much time would be needed to coördinate the material, that no attempt has here been made to do so.

17 The curve itself is not new if one considers it the same as the hydrostatic arch with the stresses all reversed, but very little, if any, study of its properties and practical application has been published. Nothing of the kind has ever come to the notice of the writer. As a matter of theoretical mechanics and even of plane geometry there seems to be an opportunity for a little addition to the technical instruction in those branches.

THE SHOCKLESS JARRING MACHINE

In this paper the following subjects have been treated:

THE SHOCKLESS JARRING MACHINE

By Wilfred Lewis, Philadelphia, Pa.

Member of the Society

The title of this paper may appear to the casual reader like a contradiction of terms, but to the foundryman who has hitherto attempted to ram sand by the jarring process, the term "shockless" will be understood to apply only to the foundation or support on which the machine stands.

2 The jarring machine is essentially a sand-packing machine, capable of ramming any mold, large or small, in a minute or less time. By the method employed, the sand is rammed, as it should be, densest at the surface of the pattern and of decreasing density above, thus favoring the escape of gases when the mold is poured. The packing of the sand results from impact between the table on which the mold is carried and the anvil on which it drops. Various means may be used to lift the table and let it drop, but in foundry work compressed air has come to be generally preferred for its convenience as a medium for the transmission of power, and also for the simplicity of the machines resulting from its use. The jarring machine is not universal in its application, nor should it be used without judgment and discrimination. Due regard must be given to the construction of the pattern so as to permit a flow of sand chiefly in one direction, and to withstand successfully the shock of impact in ramming. But, for the broad field of work adapted to its use, there is nothing comparable to the jarring machine as a saver of time and money.

DEVELOPMENT OF MACHINE MOLDING

3 Jarring machines have been in practical use for many years without attracting much attention. The records of the patent office go back to 1869; but like all other types of molding machines, they

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have never been made, until quite recently, for large, heavy work beyond the strength of one or two men to handle. It was quite natural that the molding machine should begin its development in a small way on small work; as the field has widened it has been seen that the possibilities for saving time and money in the foundry have increased with the size of the work adapted to machines.

4 Anyone who has watched a bench molder fill his little flask, ram it in a few seconds with a butt rammer in each hand, then roll it over and draw the pattern with the dexterity of an artist in legerdemain, can appreciate the difficulty of helping him in his work by any mechanical means. Yet the old-fashioned hand squeezer saved some of the ramming time, and the power squeezer saved a little more, but not so much in time as in the strength of the operator, keeping him fresh, with steady hand and eye, for the delicate work of drawing patterns and setting cores. Pattern guides were then devised to assist in drawing patterns, and vibrators were invented to free the pattern from the sand without appreciably enlarging the mold.

5 The use of molding machines on small work has resulted in a substantial saving in the cost of molding, less wear and tear on patterns, and greater uniformity in castings, the saving in weight of the castings due to the use of a vibrator being sometimes an item that soon pays for the installation of the machine.

6 The hand squeezer is, of course, limited in its application to what can be done at one effort by one man, and for larger work power squeezers have generally been employed; but when large molds are squeezed by power, more or less trouble is encountered in the distribution of pressure on the sand. At one time an effort was made to overcome this difficulty by means of a water-bag placed between the sand and the squeezing head. Better results were obtained, however, by judicious tucking in deep pockets, by heaping the sand over deep places, and by scooping it out over high points in the pattern; but the main difficulty in squeezing deep molds lies in the fact that the sand is generally moved against the pattern instead of the pattern against the sand. This results in the greatest density of sand being away from the pattern, next to the squeezing head, and not where it should be, next to the pattern.

7 As shown by Harris Tabor's experiments,[1] presented to this Society in 1892, the friction of sand on the sides of a deep flask may carry a large part of the pressure on the squeezing head. To avoid

[1] Transactions, vol. 13, p. 537.

this difficulty, bottom-ramming machines have been employed, which move the pattern against the sand, but this method contemplates a definite, predetermined movement of the pattern to produce a mold of the density desired, and is subject to variations not easily controlled. Bottom ramming has, therefore, not been adopted to any great extent, and power squeezers have usually been limited in application to flasks not more than two to three feet on a side by a foot deep. Such machines, when designed also for pattern drawing, are comparatively expensive and have marked for a time the limitations of machine molding.

8 During recent years, however, while the power squeezer and the split-pattern machine were completing their development, the much neglected jarring machine has grown steadily in favor and in size, until today there would seem to be no limit to its capacity. These machines are simple in construction and effective in operation, while on large work the saving to be effected by their use probably exceeds that by all other types of molding machines combined. I say on large work, because on small work jarring machines cannot compete with power squeezers of the same capacity, except perhaps in a few special cases where the work is deep.

VALUE OF LABOR-SAVING APPLIANCES

9 The value of any machine depends of course on what it can save, and what it costs to effect that saving; a problem to be worked out in every instance by a systematic time study of all the operations embodied in producing a given result. For instance, if it takes two men eight hours to mold by hand a certain pattern, in a flask 45 in. by 60 in. by 36 in., and if five hours of this time is consumed in ramming sand, a jarring machine would save practically five hours of the time. It would not save any of the pattern-drawing and finishing time, nor any of the time for setting cores, but it would enable two men to make the mold in three hours, instead of in eight hours by hand. Hence, with a suitable jarring machine, two men could put up 2.67 times as much work as by hand.

10 In this case the jarring machine saves more than half of the molding time, and is therefore the most important help in the reduction of cost; but when patterns are rapped with a sledge, and drawn with a crane or hoist, a great deal of the molder's time may be taken up in finishing, or, to put it in more bluntly, in repairing the damage done to the mold by this brutal way of rapping and drawing the pat-

tern. Assuming that about one hour might be spent in finishing each half-mold when made by hand, an effective pattern-drawing machine could easily save two hours; and it is evident that with such a machine two men could make a mold in six hours, thus increasing their rate of molding 1.33 times that by hand. By means of a sand conveyor, or even a clam-shell bucket on a traveling crane, perhaps 30 minutes could be saved, thus enabling two men to make the mold in $7\frac{1}{2}$ hours. With this device only, they could make their rate of molding 1.067 times as fast as by hand.

11 For the purpose of illustrating the effect of coöperation or concentrated effort upon any given piece of work, let us now assume that the demand for the castings above referred to has resulted in the making of three sets of patterns, and that we have three sets of men at work making three molds a day by hand. Now, suppose we give one set of men a jarring machine, another set a pattern-drawing machine, and the third set a sand conveyor. In 8 hours:

The first set of men will produce	2.67	molds
" second " " " " "	1.33	"
" third " " " " "	1.067	"

Six men with three patterns will produce 5.067 molds, instead of three molds by hand, or about 1.7 times as much work.

12 On the other hand, if we have but one pattern and one set of men, and give them the combined help of a jarring machine, a pattern-drawing machine and a sand conveyor, two men will save 5 hours in ramming time, 2 hours in finishing time, and half an hour in shoveling sand, or $7\frac{1}{2}$ hours in all; bringing the time on one mold down from 8 hours to 30 minutes, and increasing the production 16 times. In other words, the same assistance concentrated for the benefit of two men will result in more than three times the production, at less than one-ninth the cost per mold.

13 The above illustration shows, not only the advantage of concentrated effort in the use of labor-saving appliances, but also the wide difference in results that may be realized from the same appliances in different hands. Not only does the concentrated effort in this case save the wages of four men and produce three times as much work, but it also distributes all of the indirect charges, which must ultimately be carried by the product, over a larger output. So much for the savings to be effected.

14 On the other hand, the interest on the investment, the depreciation of the machine and the consumption of power, must be

accounted for; and in addition to all these, the damage that may be caused by the action of the jarring machine upon finished molds, or even upon buildings in the neighborhood, and the annoyance caused by noise and ground waves generally. This damage and annoyance has increased steadily with increase in the size of the machines and in the weight of the loaded table. To meet these serious objections, various expedients have been adopted, among which may be mentioned a reduction in the stroke or drop, and a more or less resilient bedding for the anvil.

PRINCIPLES GOVERNING THE DESIGN OF A JARRING MACHINE

15 These palliatives, however, left much to be desired until the shockless jarring machine, with its anvil rising up to meet the falling table was developed. This has eliminated the chief objection to jarring machines. It is the object of this paper to elucidate the principles upon which it works and to establish its claim to superior efficiency in the consumption of power.

16 In the first place, it must be admitted, that although the packing of sand by the jarring process is very quick and effective in producing results, it is not very efficient under the most favorable conditions, as far as the expenditure of power is concerned, and that under certain conditions the efficiency may be reduced to zero. In the process of ramming, the density of sand is increased 25 or 30 per cent, and if a steam indicator were attached to the cylinder of a power squeezer, it may be questioned whether it would ever show over 1000 ft. lb. per cu. ft. as the work actually done on the sand in squeezing it to proper density. Of course a great deal more power than this would be consumed in the use of water or air as a working fluid, but the work put into the sand would in all probability not exceed 1000 ft. lb. per cu. ft.

17 To produce the same effect by jarring, the sand might be raised to a height of 4 in. and dropped upon an anvil 30 times; but to the weight of the sand must be added the weight of the table and flask, and the excess sand used as an aid in ramming. The first blow struck will cause the greatest flow of sand and will do the most work upon it, while each succeeding blow will increase the density and do less and less work, until, after a certain number of blows, the sand will remain at a density corresponding to the drop. When this point has been reached, the continued action of the jarring machine simply wastes power and produces no effect. The jarring machine is therefore

more efficient during the earlier stages of the process than it can be when the condition of maximum density is approached; and for this reason, the longer the stroke the greater will be the efficiency. Other considerations of a practical nature, affecting the elasticity or durability of flasks and patterns and of the machine itself, necessarily tend to limit the stroke, however, so that in practice it varies from $\frac{3}{8}$ in. on some machines to 4 in. or more on others with an average of perhaps $2\frac{1}{2}$ in.

18 In such machines, the most important consideration is solidity of construction and freedom from vibration of the jarring table. Otherwise the sand will become broken or laminated and the mold will be liable to fall apart in handling. Although lightness of construction in the jarring table is obviously desirable from the standpoint of economy in power, it is certainly not desirable from the standpoint of making perfect molds. The good results which accompany the stronger and stiffer table really cost less and it consumes less power in the end, because there are no failures necessitating repetition, or molds to be thrown away. The importance of solidity in the jarring table will be appreciated after a consideration of the character of rammed sand. It has a certain amount of elasticity, a good deal of resistance to further compression, and some tensile strength, which of course is easily overcome. There must, therefore, be no movement between the pattern, sand and flask, tending to pull the sand apart, and of sufficient amplitude to cause fracture, and no lateral movement tending to slide one layer of sand over another. Such fracture or lamination may be caused by badly fitted pattern boards, flimsy patterns, or crooked flasks not properly bedded, but a light and flimsy jarring table that can be easily warped out of shape will augment the difficulty, and effectually prevent the success of good patterns carefully mounted in strong and firmly bedded flasks.

19 In the molding machine to be described, the table adopted has been formed in one piece with the jarring cylinder, as shown in section in Fig. 2. This table has great depth of beam, and the metal is distributed as it should be for economy of cast iron, in a broad expanse of plate on the tension side, and a smaller mass around the cylinder on the compression side, where the blow is struck. Radial ribs connect the tension and compression sides of the beam, forming a table of enormous strength and stiffness to distribute the central blow of impact equally in all directions. A table of this type is really stiffer than some of the anvils on which other tables are made to drop. At any rate, there is no perceptible vibration of the table when it strikes

its anvil, or rather the buffer ring of leather, or other non-resilient material, interposed to relieve the sharpness of the blow and to reduce the noise. This buffer also helps to reduce vibration and rebound, by reducing the intensity of the force of impact. It is not, however, the rebound of the table from its anvil that injures a mold, so much as the rebound of the flask and sand from the table. Solidity of contact between table, pattern board and flask, is one of the most important elements for the successful working of a jarring machine; yet this detail rarely receives the attention it deserves, and not unfrequently the machine is condemned for this reason, which is no fault of its own.

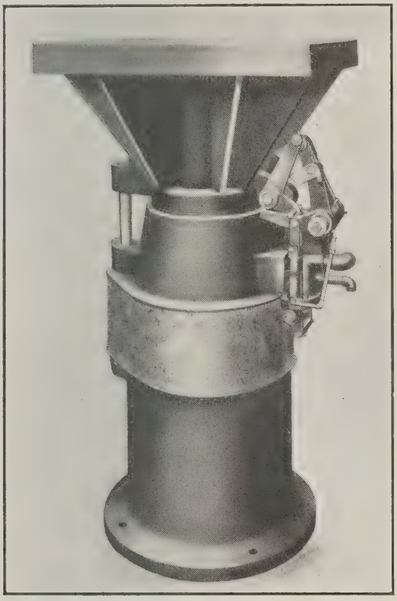

Fig. 1 Shockless Jarring Machine

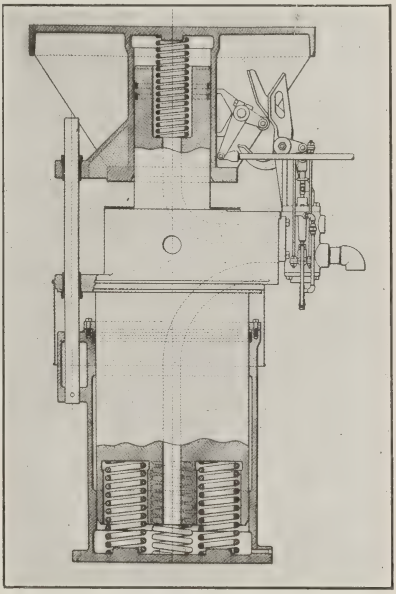

Fig. 2 Section of Jarring Machine

20 As already stated, unlimited power may be expended in jarring sand to any given density; and since there is a certain maximum density corresponding to any given drop, it is also quite evident that efficiency increases with the drop, and decreases with the dead weight handled over and above the weight of sand used. But a certain amount of dead weight is inseparable from the process, and for this reason a heavy machine may not be used to its best advantage on light work. Nevertheless, with air as a working fluid the benefit gained by expan-

sion on light work offsets to a great extent the loss from the greater proportion of dead weight carried, giving to the jarring machine which uses air expansively in its cylinder quite a wide range of capacities, under approximately uniform efficiency as far as the consumption of air per cubic foot of sand rammed is concerned.

21 But it is not only the air consumed in lifting the loaded table that may not be utilized to the best advantage. At the instant when the loaded table strikes its anvil, the sudden change in the velocity of the table, whatever that may be, measures the pressure of impact, and the ramming effect is measured by the square of that change in velocity, which is proportional to the energy absorbed, part of which is utilized in ramming sand. Therefore, the greater the change in velocity at the instant of impact the greater the ramming effect, and by the laws of impact, the heavier the anvil the better. Efficiency in a plain jarring machine naturally increases with the weight put into the anvil; but since the cost of the machine depends very largely upon the weight of cast iron or concrete used, the weight of the anvil is generally limited to that of the loaded table. When the anvil is bedded on rock, it becomes practically of infinite weight, and capable of developing the maximum ramming effect for any drop given to the table. A rock bottom does not, however, eliminate the destructive ground waves, and often facilitates their transmission to unusual distances. To mitigate the effect of these shocks, the practice has been to bed the anvil on a timber cribbing, after the manner employed for steam hammers.

EFFICIENCY DEPENDS UPON THE ANVIL

22 So cushioned, the anvil when struck by a loaded table of its own weight will suddenly acquire one-half of the velocity of the table at the instant of impact, after which both table and anvil will be brought to rest by the yielding resistance of the timber cribbing; they will then be returned by its elasticity to their normal position. The loaded table, in this case, loses at the instant of impact only one-half of the velocity it would lose by falling upon an anvil of infinite weight, as exemplified practically in an anvil founded on rock. As a result, the retardation of the table by the compression of the wooden cribbing is less intense and less effective in ramming sand, although this second change in velocity no doubt has some effect, especially in the earlier stages of the ramming process while the sand is comparatively soft. Nevertheless the initial change in velocity, between a loaded

table and a floating anvil of equal weight, is only half as great as the change in velocity of a loaded table falling the same distance upon an anvil of infinite weight; and the ramming effect in the first instance, being measured by the square of the change in velocity, is only one-quarter as much as in the second case, where the whole energy in the falling mass is immediately absorbed.

23 An anvil cushioned upon a wooden crib may be considered as a floating anvil, in which the supporting medium is very dense and highly resistant, but in which also the resistance to compression is trifling compared to that of rock. The stiffness of such an elastic bedding for an anvil might be estimated from the anvil movement, of which no data are at present available; but, however effective it may be in the initial stages of the jarring process, it can have but little, if any, effect upon the final stages after the sand has been rammed to a density in excess of that corresponding to such elastic resistance. It may be said, without hesitation, therefore, that anvils cushioned upon wooden cribbing are much less effective than anvils founded in rock, and that such anvils, equal in weight to the loaded table, have in the final stages of the jarring process a comparative efficiency of only 25 per cent.

24 In considering the mechanical efficiency of a jarring machine it is therefore a matter of some importance to provide an anvil of maximum efficiency for any given weight. As a matter of course, the heavier the anvil in any case the better, and the unit standard for all anvils may be one of infinite weight, comparable to a foundation on rock. Such an anvil stands for the highest attainable efficiency, but it is not a practicable construction on account of the destructive ground shocks, which the shockless machine eliminates. We shall presently see how the anvil in this machine compares in efficiency with the usual type of anvil mounted on a wooden crib.

GENERAL DESCRIPTION OF THE SHOCKLESS JARRING MACHINE

25 The shockless jarring machine consists, in its usual form, of a jarring table mounted upon an upstanding plunger forming the anvil, which in turn is mounted in a cylinder base and supported upon long helical steel springs. Compressed air, as the working fluid, is admitted through an automatic valve, under hand control, attached to the plunger or anvil base, and passes first into the jarring cylinder to raise the loaded table. At some predetermined point in the table movement, the air is automatically cut off from the cylinder, and

while the valve is reversing, the confined air will expand and lift the table further from its anvil, provided its initial pressure exceeds the balancing pressure due to the weight carried. Then, when the operating valve completes its reverse movement the air from the jarring cylinder may be exhausted into the atmosphere, but preferably it passes from the jarring cylinder to the anvil cylinder beneath, and the table drops by gravity against the reduced pressure in its cylinder. At the same time, the plunger base or anvil is relieved of a considerable part of the load carried by its supporting springs, which immediately expand, giving the anvil an upward velocity to meet the falling table. When air is expanded from the jarring cylinder into the anvil cylinder this upward velocity of the anvil is augmented and the falling velocity of the table is somewhat retarded, but in any case the momentum of the rising anvil is substantially equal to that of the falling table at the instant of impact. As a result, both table and anvil come to rest with great jarring or ramming effect upon the sand, but without shock or jar upon the foundation or any surrounding material.

26 When the air from the jarring cylinder is discharged at once into the atmosphere the momentum of the falling table may somewhat exceed that of the rising anvil at the instant of impact; but when this air is expanded into the anvil cylinder it compensates more or less for the loss of spring pressure as the anvil rises, and in this case the momentum of the rising anvil may exceed that of the falling table at the instant of impact. The difference, however, need not be very pronounced, and simply results in a slight initial velocity of the table and anvil at the beginning of the next stroke.

27 The advantage of the second expansion is two-fold: it utilizes the potential energy of the compressed air in augmenting the momentum of the anvil, and at the same time it checks the acceleration of the table due to gravity and holds it in contact with the load upon it while falling. Otherwise a poorly fitted pattern board or flask may tend to spring away from its support while falling and cause lost motion, productive of a bad mold. For the same purpose, when the air is discharged directly from the jarring cylinder, a long compression spring between the jarring cylinder and its plunger may be introduced with good effect. In several instances such springs as shown in Fig. 2 have been made to carry half the weight of the table with 8-in. compression. They assist in lifting the loaded table, and retard its acceleration in falling; and by their use the lifting capacity of jarring tables may be considerably augmented. Their chief pur-

pose, however, is to retard the falling table and hold the pattern flask and sand firmly against it in readiness for the coming blow. With such a spring, the action of the table is, of course, somewhat slower in falling and more stroke is required to produce a given velocity of impact. On the other hand, the table rises faster and runs further to produce a given blow, and the increased stroke reduces the percentage of clearance space to plunger displacement. The spring in this position has, therefore, some beneficial effect upon the consumption of power, while serving a much better purpose, in the production of

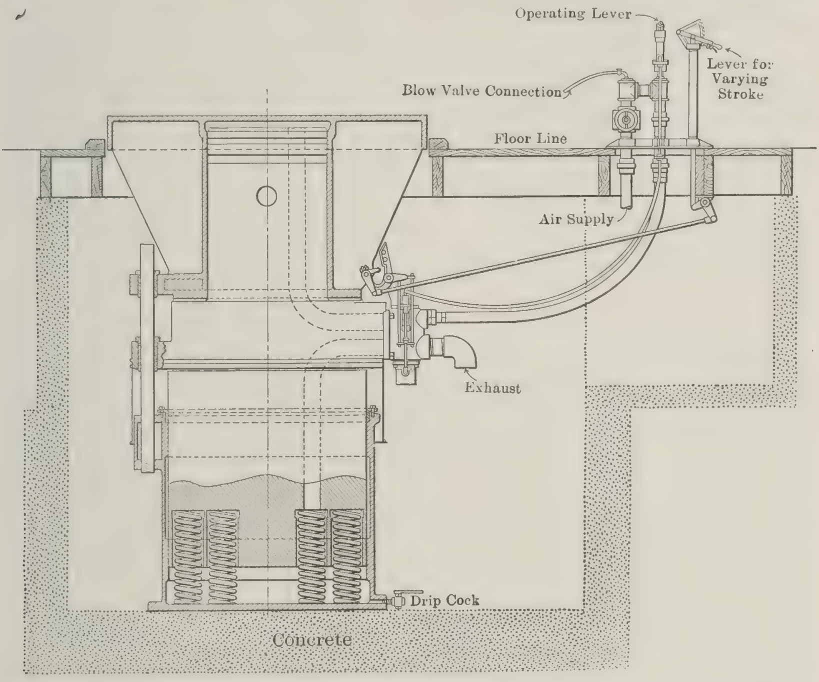

Fig 3. Section of Machine to Handle Molds Weighing 25 Tons

good molds, and although not so important when the air from the jarring cylinder passes through the anvil cylinder, it may be of some value in that case also.

28 The valve mechanism, and the means by which it is controlled, do not particularly concern the present discussion. It will suffice to say that the machine is started and stopped by an operating lever which controls the admission of air to the automatic mechanism. So long as this lever is held down, the machine will run automatic-

ally, and when released, the machine will stop. A latched lever is arranged to adjust the stroke, which can be varied while the machine is running. A safety stop is also provided to limit the table movement through the action of the main valve attached to the plunger base. When pressure is admitted to the jarring cylinder, the anvil cylinder opens to exhaust. When in action, it descends while the table is rising and then rises to meet the falling table.

29 Fig. 1 represents the design of a machine now being built for a large foundry to handle half-molds weighing 25 tons. The table is a steel casting 8 ft. x 12 ft., with lifting cylinder 3 ft. in diameter, and the plunger base forming the anvil is a solid iron casting weighing 65,000 lb. This is carried upon 22 steel springs, designed to compress 8 in. under the maximum load and to develop a working stress of only 60,000 lb. per sq. in., which is very much less than the usual working stresses on railway car springs and quite within safe limits. The total weight of the machine complete will probably be in excess of 90,000 lb. This is carried in a concrete pit designed simply to protect the machine and to support the static load on the floor of the pit.

30 The earthquake from a loaded table weighing 65,000 lb., dropping two to three inches upon an anvil bedded in the ground, can readily be imagined. Not only would it undo the work done by the machine, but a large area of valuable floor space in its vicinity would become useless and office buildings at a considerable distance might vibrate in sympathy. In this instance a comparatively small jarring machine of a well-known type, with anvil mounted on wooden cribbing had caused more or less annoyance to the occupants of office buildings in the neighborbood, and the machine above described was designed to avoid any further trouble of the same character.

31 It has been shown that a floating anvil which does not rise to meet the falling table, when equal in weight to the latter, has only one-quarter the efficiency of an anvil founded on rock. The efficiency of such an anvil, when mounted and actuated so as to acquire a momentum equal to that of the falling table at the instant of impact, remains to be determined. Obviously, the anvil will meet the table midway when the latter has fallen half the distance by which they were separated. In terms of the velocity of the table falling the whole distance, its velocity at this point will be $\sqrt{\frac{1}{2}}$, and the velocity of the anvil will be the same. But in this case the velocity of the table is entirely destroyed, while in the previous assumption the change in velocity at the instant of impact was only half the final

velocity. The relative changes in velocity are, therefore, as $\sqrt{\frac{1}{2}}$ to $\frac{1}{2}$, and the ramming effects in the two cases will be to each other as the square of $\sqrt{\frac{1}{2}}$ to the square of $\frac{1}{2}$, or as $\frac{1}{2}$ to $\frac{1}{4}$. Under the assumed conditions the anvil in the shockless machine is therefore twice as efficient as the same anvil cushioned on a wooden crib.

32 It might be shown still further that the vibratory action which develops equal momentum between the table and anvil is more efficient mechanically than any other action which develops unequal momentum. With compressed air as a working fluid, however, it pays to utilize its potential energy in the anvil cylinder rather than throw it away, and a decided gain in effect is realized in this way.

33 It may also be pointed out, that when the sand is soft the change in the velocity of the table at the instant of impact is greater than it is when rammed, because in the first condition the table movement is arrested before that of the sand, while in the latter condition, both stop together.

34 The loss of power in cylinder clearances is well known, and the obvious remedy of a short passage from the valve to the cylinder has led to the use of internal valves of more or less ingenuity and efficiency; but experience in machine design clearly points to the futility of attempting to embody all advantages or completely eliminate all disadvantages in any construction. The best machine for any purpose is the best compromise that can be made between conflicting advantages. Rather than save air by the use of a valve which is comparatively inaccessible, is it not better to sacrifice a little air for the sake of good construction, and accessibility to all working parts? At the same time, it may be said that the air consumed in the clearance passage to the jarring cylinder is not wholly wasted. It adds materially to the work done by expansion in the jarring cylinder, and again, when discharged into the anvil cylinder it adds to the momentum of the anvil. In addition to the consumption of air for any given stroke, it must not be forgotten that the blow struck in the shockless machine is twice as effective as the blow for the same expenditure of power in the usual type of jarring machine.

SPECIAL ADAPTABILITY IN HIGH BUILDINGS

35 Attention should also be called to the possibility of installing a machine of the shockless type on the upper floors of high buildings, where many foundries are now being located. The action of the machine is entirely free from jar except where it is wanted, on the

work produced, and the pulsating variation in floor load while running is no greater than is usually experienced in the operation of power squeezers. A number of these machines are now under construction for installation on upper floors, and in this connection it may be of interest to note that the original experimental machine, shown in Fig. 1, was set up on floor beams over a pit and operated without any vibration appreciable to a man standing on the beams while ramming up a half mold weighing about 1000 lb. In this case the weight of the machine was about 6000 lb., and a stroke of 4 in. was employed. As the movement of the anvil was about 1 in., it met the table when it had fallen about 3 in. The variation in the load in the floor beams was about 10 per cent of the static load carried, or between 600 and 700 lb. This variation, however, was gradual, the anvil rising and falling with the movement of the table. When impact occurred, the load on the floor beams simply ceased to decrease, and began again to increase without transmitting to the floor beams any part of the shock of impact, which was confined exclusively to the jarring table and its plunger base.

DISCUSSION ON LUBRICATION

LUBRICATION AND LUBRICANTS

By Dr. C. F. Mabery,[1] Published in The Journal for February 1910

ABSTRACT OF PAPER

A brief outline is presented of the history of lubrication, with allusion to the sources of the various lubricants. Petroleum products are briefly mentioned as to their composition and properties as lubricants. Graphite is referred to as possessing the most desirable qualities as an unctuous and extremely durable lubricant.

The use of the Carpenter frictional testing machine is explained, in testing light and heavy oils alone and with deflocculated graphite. Results are also presented on the efficiency of deflocculated graphite with water, kerosene and fuel oils as vehicles.

It appears that the coefficient of friction increases with increasing viscosity of lubricant, in close quantitative relations. The addition of 0.35% deflocculated graphite diminishes the friction of every oil and increases its efficiency and durability. Suspended in water, graphite maintains with no variation an extremely low coefficient of friction as do also kerosene and fuel oils.

Every lubricating oil has a well defined limit of load within which it can maintain a continuous film under definite conditions of oil supply and speed.

LABORATORY TESTS OF LUBRICANTS

Discussion by Dr. P. H. Conradson,[2] Franklin, Pa.

Non-Member

To make complete tests of lubricants—oils and greases—requires a great deal of expert knowledge and experience to enable the engineer to interpret correctly the results obtained. This point will be clearer perhaps, if one considers the various classes of machinery to be lubricated under all conditions of service, weather changes, and high and low temperatures.

2 The method of applying the oil to the parts to be lubricated, the condition of the bearing surfaces, composition of the journal and bearings, etc., play an important part int he proper interpretation of

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[2] Chief Chemist, Galena Signal Oil Company.

lubricating oil analysis. An oil that would give satisfaction when applied direct to the journal by means of soaked waste, might fail altogether if the wick method of feeding the oil were used; furthermore, a sight-feed cup with an orifice wide enough might give satisfaction, while a gravity feed through a long pipe of small bore might give very unsatisfactory results.

3 In making a complete investigation of the real or comparative value of a lubricating oil with another oil, we have then to consider the kind of machine to be lubricated, the service requirements and conditions, and the methods of applying the lubricant to the machine, and make our chemical or physical laboratory tests accordingly.

4 Generally speaking, the chemical tests, as made, are very inadequate, as are also the physical tests, especially frictional tests on oil-testing machines, unless the machines are constructed in such a way that the actual conditions can be approximately reproduced. For instance, in testing a spindle oil, the testing machine should be run practically at the same load and speed as the spindles are in actual service. In testing railway car and coach oils, the machine should have approximately the same size journal and bearing as would be found in actual railway car journals. The same is also true as regards speed, load, and the application of the lubricant.

5 While many valuable conclusions may be drawn from properly conducted laboratory tests, those of the greatest practical value come from a close observation of the lubricant in actual service, and we can base our laboratory investigations on the results obtained, especially in comparing different oils intended for the same work. To bring out the point more clearly, let us consider an air compressor, such as is used on street cars and on electric locomotives. As is well known, these compressors are not water-cooled or even air-cooled. It is not difficult to get an oil that will lubricate the compressor cylinder, but it is difficult to find an oil that will not carbonize at the high temperature, often 450 to 460 deg. fahr. in the street-car compressor, and 550 to 560 deg. fahr. in the electric locomotive air compressor.

6 The difficulty lies in the fact that as the compressed air passses through the outlet ports and check valves in the compressor heads, there is a very rapid increase in the temperature. The small amount of oil that goes with the compressed air, if not of the best or suitable quality will then begin to carbonize and cause trouble by forming heavy carbonaceous deposits on the check valves. Now, an ordinary oil-testing machine cannot bring out the essential requirements of a suitable oil for such service, and to make a proper laboratory investi-

gation and test of an air-compressor oil, it would be necessary to have an air compressor and test the oil as near as possible under actual service conditions.

7 Again, we may consider a steam turbine and suitable oils for its lubrication. Ordinary laboratory tests, both chemical and physical, such as are generally used in this country, do not bring out the essential qualities of a suitable turbine oil, because the service requirements and conditions are so entirely different from the general run of machinery, that special tests must be made. Therefore, from a practical point of view, to develop the essential qualities of turbine oils, it is necessary that the service conditions and requirements should be studied first, laboratory tests then being made in accordance therewith as far as practicable.

8 To illustrate, we may consider a steam turbine of the Curtis type, where the oil is forced under high pressure to the step bearing, and then returned to the oil tank. At first this might appear to be a very small matter, but in actual practice and experience it is not, for the following reasons: In the first place, leakage of steam occurs in most of these steam turbines as now constructed; this steam condenses, becomes mixed or churned in with the oil, and if the oil is not of the proper kind, it becomes emulsified. The emulsified oil gradually becomes thicker, and as the same circulating system is used for the rest of the machine, the emulsified oil often causes considerable trouble. Then again, we often find that the amount of oil used in the oil circulating system is entirely too small in quantity, so the oil has to pass through the turbine many times during the hour, in the twenty-four hours, and from week to week. This imposes a severe service requirement on the oil, which gradually becomes polymerized and oxidized, developing petroleum acids. If sulphur compounds are present to any extent, they become gradually oxidized and besides causing corrosion may cause a great deal of trouble from formation of asphaltic and tarry matter, which would clog the filters and orifices through which the oil has to pass. From a practical point of view, therefore, the laboratory tests of turbine oils should be considered along these lines. The same may be said of all lubricating oils intended for use in oil circulating systems, which are now so largely used in stationary power plants, shops and mills, war vessels, steam ships, etc.

9 It is a generally accepted idea that if the oil is adapted to the load and speed, the lower the viscosity, the better lubricant it will be. This, to my mind, holds good only where the service conditions

are uniform, and where the method of applying the oil to the bearing and journal is a positive one, such as in gravity or pressure pump systems. Where the climatic changes are great, as on railroads, this will not hold good.

10 The load and speed of the railroad trains are the same during the summer and winter, and as is well known, the practice in this country is to convey the oil to the journals by means of oil-soaked waste. A satisfactory thin winter oil with a low cold test and low viscosity, containing sufficient lubricating capacity to keep the bearing and journal apart, would not be suitable during the hot season, not because it has not the adequate sustaining power, but because of the method of applying it to the surfaces to be lubricated, making it necessary to use a much thicker oil than is theoretically required.

11 Therefore to make laboratory tests of the relative lubricating values of oils considered from a practical standpoint and to draw correct conclusions from the results obtained, we must consider the kind of machine or machines to be lubricated; the speed and the load; the composition of the bearing metal; whether the journals are iron or hard steel; the method of applying the lubricant, either with wick feed, soaked waste, sight-feed cups, flooded bearing or continuous oil-circulating system; the actual service requirements and the climatic conditions. We must make complete chemical and physical tests as near as possible in accordance with these conditions. I might with propriety state that one oil can not be considered a better lubricant than another oil unless the service conditions and requirements are specified and fully understood and the laboratory tests made in accordance therewith.

12 On the diagrams presented by Professor Mabery the first point of great interest seen is that his tests are run say, for two hours, to get all conditions uniform with a constant and given feed of oil; then the oil supply is shut off without stopping the machine, the tests being continued until the friction and temperature begin to rise rapidly, the time of the break being noted. The length of time from the shutting off of the oil supply to the break would, according to Professor Mabery's theory indicate the comparative lubricating value or endurance of the two oils, other things being equal; that is, if we take two oils intended for the same machinery and same service requirements and conditions, the oil that runs longer on the test machine after the supply is shut off, would be considered the better oil. I think that investigations can be made to great advantage in estimating the comparative lubricating value of two or more oils, by varying

the load and speed to a greater extent than Professor Mabery has done, as well as keeping the temperature of bearing and journal constant during the tests.

13 The second point of interest in Professor Mabery's results is the introduction of a very small amount of pure colloidal graphite in the oil to be tested. Especially interesting are the curves obtained from the mixture of water and graphite.

14 Professor Mabery called attention to the fact that crude oils from different sources as well as different treatment in the refining, produce lubricating oils of great variety, not only as regards the chemical composition and physical qualities, but also in lubricating value. Table 1 is of interest in this connection. Nos. 1, 2, 3, 4 and 5 are the same oil fractionated by means of "Florida Earth." While the flash and fire tests and the gravity remain practically the same, with changes in color and some also in viscosity, the congealing point or cold test has changed materially from the original No. 1, during the successive stages. From a technical and practical standpoint it would indeed be of great value, if Professor Mabery would make frictional machine tests (endurance tests) of these oils, to determine their relative lubricating values.

TABLE 1 LUBRICATING OILS

	1	2	3	4	5	6	7
Flash Point, °F.	365	370	370	370	370	410	415
Burning Point, °F.	440	440	440	440	440	470	480
Gravity, 60°F.	30	30.7	30.5	30.3	30.2	23.0	22.8
Color, °Beaumé.	Very dark	Yellowish	Orange	Red	Deep red	Red	Dark red
Cold Test, °F.	Zero	+18	+25	+25	+25	+5	+20
Viscosity (P.R.R.) pipette							
@ 125° F.	98	92	94	94	93	85	162
100°	160	149	148	153	151	143	309
90°	205	184	192	195	195	188	410
80°	271	238	249	254	254	253	Drops

15 Nos. 6 and 7 are the same oil before and after being in continuous service in an oil-circulating system about ten months. Here again the flash, fire and gravity tests remain practically the same. The color, as would be expected, has grown somewhat darker, but the congealing point and body or viscosity have greatly changed. The fresh oil had sufficient body or viscosity and gave satisfaction in service at the start, but its durability (endurance) in actual service, as seen from

the rapidly increasing sluggishness or viscosity, is seriously questioned Would Professor Mabery's endurance frictional tests, such as he conducted, have brought out the lack of stability of this oil? This is a practical question that practical men want to know, and to make

TABLE 2 TESTS ON GALENA SUMMER OIL

Bearing metal, brass; journal, steel; bearing surface: length 3.9 in., diameter 3.75 in., width 1.9 in., area 7.4 in.

Number of Test	1	2	3	4	5	6	7	8	9
Pressure on Journal, lbs. total.	1000	2000	3000	3500	4000	5000	5500	6000	6500
Pressure on Journal, lb. per sq. in.	135	270	405	473	540	675	743	810	878
Method of lubrication, flooding bearing.									
Minimum Coefficient of Friction.	020	019	0163	016	0156	0158	0153	0152	0151
Maximum Temperature of Journal, °F.	109	114	115	118	122	134	136	140	144
Temperature of Room, °F.	68	68	68	69	68	70	71	71	71
Elevation Temperature Journal above room	41	46	47	49	54	64	65	69	73
R. p. m. of Journal.	215	220	220	220	223	220	220	220	185
Feet traveled by rubbing surface per. min.	211	216	216	216	219	216	216	216	182

No. of Test	Time	R.p.m.	Temp. Journal °F.	Total Friction Lbs.	Coefficient Friction	No. of Test	Time	R.p.m.	Temp. Journal °F.	Total Friction Lbs.	Coefficient Friction
1	8.40	210	106	20	0.020	6	10.50	220	133	79.1	0.0458
	45	218	107	20	0.020		55	220	134	79.1	0.0158
	50	216	109	20	0.020		11.00	220	134	79.1	0.0158
2	9.00	220	112	38	0.019	7	11.10	220	135	84.0	0.0153
	05	222	114	38	0.019		15	219	136	84.0	0.0153
	10	218	114	38	0.019		20	221	136	84.0	0.0153
3	9.20	220	114	49	0.0163	8	11.30	220	138	92.0	0.0153
	25	220	114	49	0.0163		35	218	140	92.0	0.0153
	30	220	117	49	0.0163		40	221	140	92.0	0.0153
4	9.40	220	117	56	0.016	9	11.50	214	142	104.0	0.0160
	45	220	117	56	0.016		55	180	144	104.0	0.0160
	50	220	118	56	0.016		58	160	145	104.0	0.0160
5	10.00	216	120	63	0.0157	10*					
	05	218	121	63	0.0157						
	10	234	122	63	0.0157						

*Journal stopped at 11.58; pressure 6500 lb.

laboratory tests of real, practical value, problems of this nature must be satisfactorily worked out and answered.

16 In Professor Mabery's endurance tests on the Carpenter machine, only 150 lb. pressure per sq. in. was used on the test journal.

Table 2 gives results of tests of a Galena railway summer oil on the Carpenter oil-testing machine at Cornell University, with increasing loads on the journal from 135 lb. to 878 lb. per sq. in.; also a sort of endurance test to determine the load capacity of the oil.

17 From the results obtained, some interesting curves as to friction and temperatures could be plotted. The main point, however, is to show that with the Galena oil, after the pressure on the journal rises above 400 lb., and up to the maximum load used in the test, the coefficient of friction remains practically stationary, and would give a nearly horizontal curve From a practical standpoint this information is of great value.

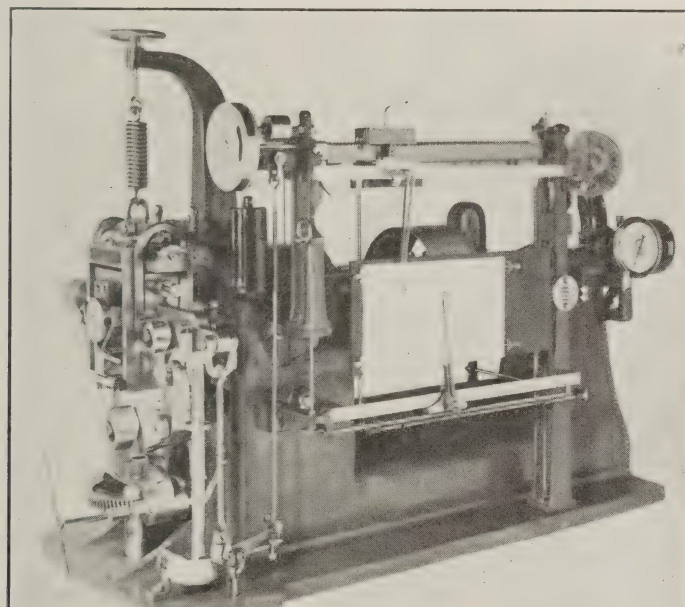


Fig. 1 Oil Testing Machine, 20,000 lb. capacity for bearings up to 5 in. by 9 in.

18 In connection with the foregoing a few comparative frictional oil tests made on the Galena-Signal Oil Company's oil testing machine will be of interest, because it is the largest and most complete oil-testing machine ever built. Full size M. C. B. car journal boxes and bearings can be inserted. By means of a system of levers, with screw and spring balance, varying loads up to 20,000 lb. can be applied on the test journal, while the machine is running in either direction at

any desired speed up to its maximum. The weight of the load is indicated on a dial; and the friction on the periphery of the journal, in pounds, is recorded on the scale beam in front of the machine. There is also a temperature indicator, a revolution counter and tacho-

TABLE 3 CONSTANT-TEMPERATURE TESTS

Comparative Frictional Tests between Pure Rape-Seed Oil and Winter Galena Railroad Car Oil (Zero Cold Test Oil)

Steel journal, size 5 in. by 9 in. bearing, genuine babbitt; total load on bearing, 10,000 lb.; projected area 15.5 sq. in.; area of contact = 16.40 sq. in.

Pressure per sq. in. projected area..................645 lb.
Manner of lubrication............................ .oil bath
Average friction of four tests for each temperature

Rape-Seed oil			Winter Galena Car Oil		
Temperature Deg. F.	Total Friction Lbs.	Coefficient of Friction	Temperature Deg. F.	Total Friction Lbs.	Coefficient of Friction
300 r.p.m.=392.5 ft. surface speed					
50	21.06	0.00211	50	20.37	0.00205
70	15.375	0.00154	70	14.54	0.00146
90	13.875	0.00139	90	12.75	0.00128
600 r.p.m.=785 ft. surface speed					
56	26.250	0.00263	60	19.94	0.00199
70	25.375	0.00254	70	18.31	0.00183
90	19.75	0.00198	90	15.31	0.00153
Viscosity P. R. R.					
@ 125°F.	80 Units			72 Units	
100°	125 "			104 "	
90°	141 "			125 "	
80°	186 "			160 "	
Cold Test	+15°F.			—5°F.	

By constant temperature is meant that the oil-bath and bearing is kept at uniform constant temperature during the whole time of test, which lasts not less than one hour after the desired constant temperature of oil and bearing is reached.

meter, and an autographic arrangement showing the friction corresponding to the number of turns of the machine. The test journals and bearings are provided with a device for passing through either water or steam during the tests and there are arrangements for applying the lubricant by any desired method during the tests.

19 In Table 3 we find, first, three series of tests at constant but different temperatures; second, two series of great difference in speed (300 and 600 r. p. m.), all other things being the same; third, compara-

TABLE 4 CONSTANT-TEMPERATURE TESTS

COMPARATIVE FRICTIONAL TESTS BETWEEN WINTER AND SUMMER GALENA RAILROAD CAR OIL

Steel journal 5 in. by 9 in.; bronze bearing: 7800 lb. total load on bearing; 300 lb. pressure per sq. in.; 27.7 sq. in. area of contact; 363 r. p. m. 475 ft. surface speed. Manner of lubrication: oil bath. Average friction of four tests for each temperature.

	Galena Winter Car Oil		Galena Summer Car Oil	
Temperature 65°F.				
Friction, right, lbs.	18.50	18.50	41.00	41.00
Friction, left, lbs.	18.50	18.50	41.50	41.50
Friction, average, lbs.	18.50		41.25	
Coefficient of friction	0.00237		0.00529	
Mean resistance per sq. in. of surface	0.665		1.489	
Temperature 80°F.				
Friction, right, lbs.	15.50	15.00	29.50	28.50
Friction, left, lbs.	15.00	15.00	30.25	29.50
Friction, average, lbs.	15.125		29.44	
Coefficient of friction	0.00196		0.00377	
Mean resistance per sq. in. of surface, lbs.	0.546		1.063	
Temperature 100°F.				
Friction, right, lbs.	11.25	11.00	20.00	20.00
Friction, left, lbs.	10.00	10.00	20.00	20.00
Friction, average, lbs.	10.563		20.063	
Coefficient of friction	0.00135		0.00257	
Mean resistance per sq. in. of surface, lbs.	0.382		0.724	
Flashing Point, deg. F.	315°F.		395°F.	
Burning Point, deg. F.	370°F.		455°F.	
Gravity at 60 deg. F.	27.3°Beaumé		24.3°B.	
Cold Test	+2°F.		+36°F.	
Viscosity (P. R. R.)				
Pipette, 125°F.	83 Units		189 Units	
100°	124 "		343 "	
90°	156 "		473 "	
80°	201 "		—	

The mean resistance per square inch of surface is obtained by dividing the average total friction by the number of square inches (27.7) area of contact. See Note to Table 3.

tive tests of a purely vegetable oil (rape-seed) with a compounded petroleum oil (Galena lead-oxide process) with viscosities not far apart, as measured with the pipette viscosimeter; fourth, the difference in

friction of the two oils at the slower speed (300 r. p. m.) is very small, while the difference at the greater speed (600 r. p. m.) is considerable, the Galena oil having a much lower friction. This is very instructive when we consider the nature of these two oils, as well as the great load and speed.

20 Table 4 gives an interesting comparison between a winter and summer Galena car oil. Both have sufficient body to carry the heaviest load and speed in railroad service, but owing to the present method

TABLE 5 GALENA CAR, COACH AND ENGINE OILS

Showing Wide Range of Viscosity and Cold Test

Dudley viscosity pipette, 100 cu. cm. water at 60°F. (15.5°C.) 32 sec.
Redwood viscosimeter, 50 cu. cm. rape-seed oil 60°F. (15.5°C.) 535 sec.

Viscosity taken at 100°F., 37.7°Cel.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Dudley	104	125	140	162	177	195	220	320	276	300	252	348	375	401	426
Redwood	151	183	204	240	260	283	335	378	392	444	480	510	556	595	650

Viscosity taken with Dr. Dudley viscosimeter–100 cu. cm. water at 60°F. (15.5°C.) 32 sec.
Instrument kept in air bath at same temperature as the oils.

Time in Seconds

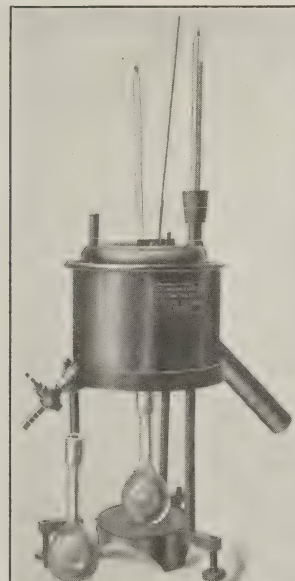
125°F. (51.6°C.)	67	71	86	109	125	138	150	171	197
100° (37.7°C.)	96	99	132	185	214	247	277	318	370
90° (32.2°C.)	115	124	170	243	291	321	362	422	509
80° (26.6°C.)	141	156	216	318	380	439	501		

	Flash Point Open Cup	Burning Point	Gravity at 60°F. Beaumé	Cold Test
Summer				
Car	350°–380°F.	425°–450°F.	26.1 to 27.7°	+20° to +40°F.
Engine	350°–380°F.	425°–450°F.	25.4 to 26.4°	+20° to +40°F.
Coach	350°–380°F.	425°–450°F.	24.3 to 24.6°	+20° to +40°F.
Winter				
Car	210°–300°F.	260°–380°F.	27.4 to 29.0°	–5° to +10°F.
Engine	210°–300°F.	260°–380°F.	26.6 to 27.9°	–5° to +10°F.
Coach	210°–300°F.	260°–380°F.	25.5 to 26.4°	–5° to +10°F.

of conveying oil to the car journals, a thick and sluggish oil with unnecessarily high viscosity is or has to be used during the warm or hot weather, naturally increasing the total train journal resistance, which of course, means excessive coal consumption. Note the great difference in the frictional resistance between these oils. Practical railroaders should ponder a little more on these facts and utilize such knowledge.

21 In connection with preceding tables, Table 5 is of interest. It gives a comparison between two viscosimeters, and illustrates the wide variation in congealing points or cold tests and viscosity possessed by first-class railroad lubricating oils, suitable to all services and climatic conditions. A close study of Tables 5 and 6 will be of great assistance to the practical user of lubricants.

22 I have spoken of the importance of adequate chemical tests in connection with physical and frictional tests. The following tests are therefore useful. While in some cases it is not necessary to subject



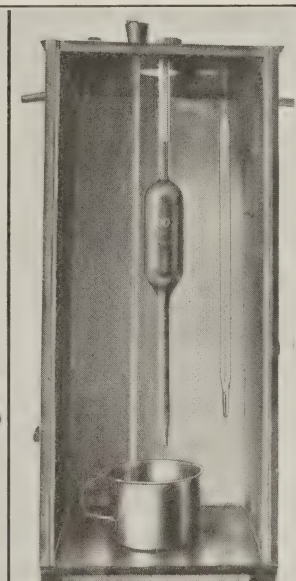


FIG. 2
REDWOOD VISCOSIMETER

FIG. 3
DUDLEY PIPETTE VISCOSIMETER

IN USING THE DUDLEY PIPETTE IT IS PLACED IN A CLOSED BOX WITH GLASS DOOR AND THE TEMPERATURE IN THE BOX IS KEPT CONSTANT

the oil to all these tests, they are of great importance in connection with special or unusual service requirements and conditions.

23 *Flash Point and Burning Point.* This test indicates the temperature at which the more volatile elements in the oil begin to vaporize to such an extent as to flash when a small flame is moved over and near the surface of the heated oil. The so-called open-cup method is generally used, and the heat is continued till the oil begins to burn when the test flame is applied. A flash and fire test that is too low

may be objectionable on account of danger from fire, besides causing too large a loss from evaporation under given service conditions. In connection with the viscosity, congealing point or cold test and gravity, the flash and fire test also enables the analyst to form an idea of the source of the petroleum.

24 *Gravity.* The test for gravity is a complementary test which enables the analyst in many cases to form an idea whether the oil is a Pennsylvania, Virginia or Western oil, the last generally having a much higher gravity.

25 *Color.* While not of much importance, other things being equal, an oil with a lighter color or pleasing appearance is oftentimes preferred to a very dark-colored oil.

26 *Odor* at times aids in detecting the kind and quality of fat oils in compounded oils.

27 *Purity* is freedom from water or matters in suspension, whether the oil is clear or turbid, etc.

28 *Gasoline Test.* This test indicates the presence of other foreign matter, or tar and asphaltic matter, if the oil gives a clear solution with 88 deg. gasoline before it is heated to burning point, but gives a precipitate with 88 deg. gasoline after this test. This indicates petroleum compounds which are readily acted upon by heat. Such an oil, in comparison with another oil, other things being equal, would not have the same lubricating value.

29 *Cloud Test* of a lubricating oil is sometimes of value in determining the amount of paraffines present and the behavior of the oil in chilling down to a temperature above congealing point.

30 *Cold Test--Congealing Point--Melting Point.* This test, together with comparative tests of the viscosity of the oil, is of much importance in connection with the service ability of the oil; and should be given careful consideration. The method used in determining the cold test and melting point, as well as getting at the comparative fluidity or sluggishness at temperatures lower than 70 deg. fahr., is not generally considered as it should be. To illustrate:

31 By the so-called P. R. R. method of cold test, valve or cylinder oil is frozen direct in an ice mixture and then stirred by the thermometer till it begins to flow when the bottle is inverted. By this method the oil may show a cold test of say 40 deg. fahr., and if no further observation is taken one has no idea of the fluidity at, say, 60 to 70 deg. fahr.; that is, the cold test would give 40 deg. fahr., but in reality the oil at 60 deg. fahr. would be so sluggish that it would hardly feed through the narrow bore feed pipe to the steam chest and

cylinders, and unless the analyst knows for a fact that the engineer has his pipe covered or warmed in some way, trouble may arise and the oil will be condemned, though it might be of the best quality.

32 Again, if the congealing point is taken by the Standard Oil Co.'s method, that is, the oil with the thermometer inserted in the bottle is put into a cooling box and gradually cooled till the oil just ceases to flow, then the bottle is inverted, or still better, the thermometer stem is lifted from time to time and note is made when the oil "hardly flows" from the stem. Without further observation however, this method,like the P. R. R. method, does not tell all. The rate of cooling or chilling—the time the oil remains in the chilling or cold-test box—plays an important part in proper interpretation of the comparative value of the oil in actual service.

33 *Viscosity.* This test is the bugbear of the oil tester—it may mean so much or so little. Certainly, in my work with the cold test and flash point, one might have a good idea of the quality and adaptability of the oil with the aid of a good viscosimeter.

34 As an adjunct to other tests, by careful study and knowledge of the service requirements, I have found the viscosity tests of the utmost value. In fact from my knowledge of all the analytical data, with the aid of the viscosimeter I can predict quite accurately the comparative friction of two oils under given conditions.

35 *Microscopic Tests.* In testing dark-colored oils, heavy machine oils, and cylinder oils, it is well to put a few drops on a slide and examine under the miscroscope. If carbonaceous matter is held in suspension, or if paraffine crystals are present at ordinary temperatures, by warming the oil a little the paraffine can be made to disappear, and other foreign matters held in suspension are brought out. Other things being equal, an oil that is free or practically so from black carbonaceous specks or flakes is certainly superior to an oil containing these in some quantity.

36 *Saponifiable Fats.* I will not discuss the method to determine these, but will merely point out that two cylinder oils, one containing thirty per cent of fat oil, and the other only 10 to 20 per cent of fat oil, other things being equal, while not of the same intrinsic commercial value, may have an equally good lubricating value. Again, a cylinder oil containing twenty-five to thirty per cent of good fat oil, might give excellent results in a steam engine at 100 to 150 lb. pressure per sq. in., the exhaust steam not being condensed or used over again in the boiler. Yet this oil might be very objectionable in connection with superheated steam and a surface condenser, the reason of course being obvious.

37 *Free Fatty Acids.* Other things being equal, the less free fatty acids are present the better.

38 *Petroleum Acids, Sulphuric Acids, Chemicals.* The presence of pertoleum acids, sulphuric acid, sulphonates and chemicals from imperfectly refined petroleum oils should always be carefully investigated, as the presence of these foreign materials in a lubricating oil, at least for certain services, might lead to serious trouble and complications. A first-class lubrication oil should be free from, or at least contain only traces of these impurities.

39 *Sulphur.* In general very little attention is paid to sulphur and organic sulphur compounds that may be present in lubricating oils. In the future, the sulphur in lubricating oils will have to be reckoned with when these oils are used for turbine service, or where the oil is used over and over again as in an oil-circulating system. Under these conditions the oil, due to the continuous exposure to heat, air, moisture and metal bearings, gradually becomes oxidized and polymerized, forming acid petroleum products, which change it both chemically and physically. The sulphur compounds present in the oil largely augment the corrosive or pitting action on the bearings and journals.

40 In the examination of lubricating oils for sulphur contents, it is important to make a distinction as to how they occur in the oil. I have often found that in making sulphur tests by burning a given amount of oil in lamps to take up the products of combustion in a carbonate of soda solution, it is necessary to consume all the oil in the test lamps, and to make a determination of the sulphur compounds left in the wick. In some poorly chemically refined lubricating oils, the sulphur compounds found in the wick oftentimes amount to from twenty to forty per cent of the total sulphur present.

41 *Maumene Test.* This is the sulphuric acid thermal test, and is of value in connection with tests of compounded lubricating oils.

42 *Evaporating Tests.* By exposing the oils in shallow flat-bottomed dishes in an air bath at 212 to 300 deg. fahr. for six hours, noting the percentage of loss and condition of residue and its behavior when mixed with 88 deg. gasoline, we obtain valuable information as to the amount of volatile matter at low temperatures.

43 *Heat Tests.* For certain service such as air compressors not water-cooled, turbines, etc., valuable data may be obtained by exposing the oils in shallow flat-bottomed dishes in a covered air bath through which air is blown for six hours, at temperatures of 425 to 450 deg. fahr.; studying the residue in the dish by dissolving the same in 88 deg. gasoline, noting whether the gasoline solution is clear or turbid, and the amount of precipitate, if any, on standing.

44 *Emulsifying Tests.* To determine the adaptability of an oil for lubrication in turbines of the Curtis type (step-bearing) it is of the utmost importance to ascertain the behavior of the oil when coming in contact with steam through the step-bearings, whether it forms a thick creamy emulsion or separates readily from the steam and condensed water.

45 *Tabulation of Chemical Tests:*

Flash point
Burning point
Gravity
Color
Odor
Purity
Gasolene tests, before and after flash
Cloud test
Cold test
Viscosity
Microscopic test for carbonaceous matter in suspension
Saponifiable fats
Free fatty acids
Petroleum acids
Sulphuric acid
Chemicals from imperfect refining
Sulphur-lamp test and in wick
Maumene test
Iodine test

Evaporating tests, a given time at 200 to 300 deg. fahr. to study percentage of volatile, and behavior of residues in 88 deg. gasolene tests and acidity.
Heat test, in air bath blowing air over the oil at 425 deg. fahr. and 540 deg. fahr. Examination of residue.
Emulsifying tests, to determine adaptability of the oil, say in turbine service
Tar and coke-forming elements present before and after heat test
Oxidation or gumming tests.
Superheated steam tests
Carbonizing tests in connection with air compressor (not water-cooled) automobile gas-engine lubrication
Capillarity or wick tests.

46 *Frictional tests.* To make frictional tests of oils and greases of practical value requires testing machines constructed for various loads, speeds, sizes of journal and bearings, and methods of applying the lubricant, comparable to those in actual service, as well as devices to keep journal and bearing at any desired constant temperature during the tests.

47 The constant-temperature tests are of importance not only for the purpose of standardizing the machine to get all conditions of bearing, journal and feed, properly regulated before the actual tests begin, but equally so for making comparative frictional tests of two oils of approximately the same viscosity. The two oils may show practically the same friction at a given temperature, but to keep the journal and bearing at this temperature, one may require a great deal more water or steam passing through the journal and bearing; again the friction may be practically the same at a temperature

of say 150 or 125 deg. fahr., but very different at 90 or at 70 deg. fahr. The constant-temperature frictional tests are therefore of great value in conducting comparative tests.

48 As a rule the reports of frictional tests are very incomplete; they should give all the constants and data taken, such as area of contact, projected area, total pressure on journal in pounds, pressure per square inch in pounds, total maximum, minimum and average friction in pounds, coefficient of friction, temperature of journal and bearing, number of revolutions and feet traveled by rubbing surface per minute, duration of tests, constant or freely increasing temperature, amount of lubricant and method of feed, besides complete analytical chemical data.

49 Where the service conditions are uniform or fairly constant, as in mills or power houses, the comparative viscosity and the congealing or fluidity points and friction, other things being equal, would determine the most economical or suitable oil or grease for the service. But in making comparative tests, chemical, physical and frictional, of lubricants under varying service conditions, expecially climatic conditions, it should be borne in mind that two oils showing considerable difference in viscosity, congealing or fluidity, and friction, may be equally good for the service requirements. Manufacturers should submit for comparative tests, samples intended to do service not for the whole year but for the different seasons.

50 In conclusion, what function should a lubricant perform? What are the necessary requisites or qualities that should be inherent in a first-class lubricating oil? These are trite questions, and will be answered briefly:

51 First, the function of a lubricant is to keep the rubbing surfaces (journal and bearing) apart to prevent undue abrasion, friction and heating.

52 Secondly, the necessary requisites or qualities that a first-class lubricating oil should possess in a high degree may be enumerated as follows: Necessary body to withstand the severest pressure in the service for which the oil is intended, so as to keep the rubbing surfaces apart, forming a continuous film between them, filling up the inequalities in the surfaces; the quality of spreading itself rapidly over the rubbing surfaces, with the requisite degree of adhesive power to remain between the rubbing surfaces without creating undue friction and heating; requisite mobility or fluidity at all seasons of the year, and in all climates from the coldest to the hottest, without impairment of the necessary intrinsic lubricating body for the required service;

durability; freedom from mineral and organic impurities, tarry and asphaltic matters; non-drying, non-gumming.

53 Yet no matter how excellent and suitable a material or machine may be, if it is not properly applied or used, the best and most economical results are not obtained. This has brought about the idea of the oil manufacturer's employing practical and experienced men educated in actual service to follow up and watch the proper application and economic use of the various lubricating oils and greases. These men have demonstrated, not only to their employers, but also to the consumer, the practical and economic value of their educational work.

54 The importance of this "following up" is far-reaching. It has gradually brought about a much more systematic and uniform method in lubrication; it has brought about greater economy in the consumption of lubricating oils, and at the same time demonstrated the possibility of better lubrication. In fact, in many instances the consumption of oil has been reduced from 50 to 100 per cent, without impairment of the best and most economic lubrication.

55 From these remarks, you will readily appreciate that to make laboratory tests of lubricants of real practical value, not only to the consumer but also to the manufacturer, involves considerable technical and practical knowledge and experience, besides full and complete laboratory equipment and experience; and the chemist or engineer who is called upon to give a qualified opinion as to the relative, comparative lubricating values of two oils or greases for a given service, considered from a practical and economical service standpoint, has indeed a difficult and oftentimes thankless task to perform.

FURTHER DISCUSSION ON LUBRICATION

William M. Davis.[1] I have not had very much experience with oil-testing machines, preferring to determine the lubricating value of two oils by actual use on the engines or machinery, assuming, of course, that the oil which will keep the bearing cooler is the better lubricant. In my capacity as oil inspector for a corporation in the New England States operating a large number of mills, I found that in some of the mills a heavy engine oil, compounded with about 10 per cent of lard oil, was used on their engines, while the manufacturing department used an excellent grade of straight petroleum machine oil.

[1] Lubrication Engineer, 93 Broad Street, Boston.

2 It occurred to me that the machine oil, which cost about ten cents a gallon less than the engine oil, was quite good enough to use on the engines. One objection to the compounded oil, however, was that it caused gumming in the cups, and sometimes made it difficult to clean the generator and motor coils when gummed on them.

3 In one engine room a thermometer placed in the main bearing of the engine while using the engine oil showed a maximum temperature of 135 deg., the temperature of the room being 80 deg., a rise of temperature due to friction of 55 deg. Readings were taking hourly from the time the engine started in the morning until shutting down at 6 p.m.

4 A bearing under ordinary running conditions will reach a point where the temperature remains constant, the heat radiated equaling the heat generated, as long as the oil feed, the speed and the pressure or load are constant. From the results of the test, Curve A, Fig. 1,

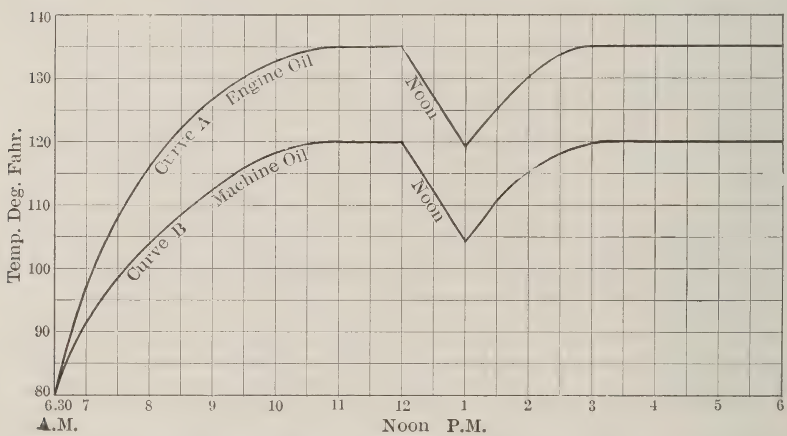


FIG. 1 CURVES PLOTTED FROM TESTS ON OILS

was plotted. As the engine was shut down for forty-five minutes at noon the bearings cooled off a few degrees, but reached the maximum temperature again in an hour or so and remained constant the rest of the day. The temperature of the room remained constant at about 80 deg.

5 The next day the test was repeated, using the machine oil. As will be seen by Curve B, the maximum temperature was only 120 deg., the temperature of the room remaining at 80 deg., showing a rise of temperature due to friction of only 40 deg. This test was repeated on several different engines at different times and the results

were practically the same in each case, proving conclusively that the machine oil was a better lubricant.

6 I do not attribute the higher temperature while using the engine oil wholly to the fact that it contained lard oil, but to the fact that it was considerably higher in viscosity than the machine oil. The viscosity of the engine oil (by Saybolt viscosimeter) was 240 seconds at 100 deg. fahr., and that of the machine oil 178 seconds at 100 deg. I prefer to make the tests in this way, for as Professor Mabery and Dr. Conradson have said, the results of tests on friction-testing machines may not agree with results obtained in actual service.

7 Another test under somewhat similar but more severe conditions showed that the straight petroleum oil kept the bearings cooler than the compounded oil. In this case, however, the oils were of the same viscosity and flash tests. The tests were made on the 22-in. main bearings of a large compound engine making about 120 r.p.m.—a very high speed for a shaft of this size. The engine was fitted with a continuous oiling system, the oil draining back into a chamber under the bearings, no cooling coils being provided to reduce the temperature of the oil. The temperature of the oils flowing on the bearings was 130 deg.

8 On August 25, while using the compounded oil, the maximum temperature of the bearing was 170 deg.—a rise of temperature due to friction of 67 deg., the temperature of the room being 103 deg. The next day while using the straight petroleum oil the maximum temperature was 148 deg., and the following day it was 138 deg., the temperature of the room being 88 deg.,—a rise of temperature due to friction of 54 deg. and 56 deg. respectively. These results showed that the petroleum oil kept the bearings cooler than the compounded oil.

Henry Souther. In the evolution of the automobile business I was called upon to approach the problem of lubricating the automobile engine. The needs are peculiar and absolutely different from those of the steam engine, and very different from those of the large gas engine. I went at the problem in the belief that as high temperatures were to be met, high flash points would be necessary. I very soon discovered that high flash points meant residue oils of high viscosity, and that these oils gummed in the cylinders to such an extent that although they lubricated the engine fairly well while running, the engine very soon refused to run because clogged with tarry matter.

2 A lighter oil was then tried in the cylinders, and a heavy oil in

the crank bearings. These worked well enough until the two oils began to mix. The oil resulting was better than the thick oil formerly used, but still unsatisfactory. I then proceeded to obtain a large variety of oils and try them out. The equipment of my laboratory included a four-cylinder engine running against a dynamo brake; and I had the use of several automobiles. I knew from experience that one of them was easy to lubricate, while another was exceptionally difficult. The engine of the latter was very sensitive, being of very high compression and very close in all its clearances.

3 I soon learned that oils that gave good practical results in an automobile engine possessed certain characteristics; not those obtained with an oil-testing machine, but obtainable with a viscosimeter, with flash and fire tests and with the Westphal balance (specific gravity). But even these tests did not seem to tell the whole story. I had read more or less about certain carbon-residue tests, but rather doubted the efficacy of any such test and the possibility of duplicating any results by destructive distillation of the oil. Nevertheless, I gave instructions to a laboratory assistant, and though he was skeptical at first, he reported in two or three weeks that he thought he could duplicate the results and prove it to me. I checked him by giving him unknown samples and duplicate samples, and he obtained results ranging from 0.1 per cent to 1.5 per cent total residue and checked closely.

4 Thus from these tests I learned what I still consider the desirable characteristics of an automobile-engine lubricant. This is a lubricant which will take care of the main shaft bearings, of the crank-pin bearings and of the piston, and one which when it passes up into the explosion chamber will disappear as much as possible; that is, will leave as little residue as possible. Whether the residue is carbonaceous or not I do not pretend to say; it looks like carbon in a coke-like condition. However, a good oil, as judged by these characteristics, will leave less residue than a bad oil.

5 I measured the viscosity both hot (210 deg. fahr.) and cold (70 deg fahr.). The viscosity when hot, measured by the Saybolt viscosimeter, should be between 40 and 50 seconds. Tested cold (70 deg. fahr.) it should not be over 300 seconds. This latter viscosity limit is placed to exclude the use of residue oils. No distillate oil can be made that will exceed 300 seconds viscosity at 70 deg. fahr., so that my specifications exclude absolutely a residue oil and accept a distillate oil.

6 As to the specific gravity there is some doubt. I am inclined to favor a high Beaumé test—between 28 and 31.

7 In the carbon-residue test I have set an upper limit of 0.5 per cent. In this test the personal equation is admittedly considerable, as is also the fact with many other tests of a practical nature. A number of assistants in my laboratory have been so trained that they can duplicate each other's results without trouble. I believe that other chemists or other testers can duplicate results by following the written instructions for this test, which is more than can be done with some chemical operations.

8 However, tests of the kind referred to in Professor Mabery's diagrams are absolutely valueless in connection with the lubrication of an automobile engine, in which the main crank bearings, the pins and the main journals are flooded with oil.

9 The popular method of lubrication, and I think perhaps the most successful, is the circulating system, which keeps the oil flowing to and from the bearings. The only objection to all systems is that the oil will work into the explosion chamber and leave a coke-like deposit or residue, which when heated to incandescence causes premature ignition. In that case smooth running is impossible. I think that a high compression engine is much more sensitive to this trouble than a low-compression engine. With the former type the temperatures seem to be so much higher that the so-called carbon deposit forms much harder and remains incandescent longer. That may not be true, but it seems to be true from my experience.

Fred R. Low. Some months ago *Power* was asked how much oil was ordinarily used to lubricate the cylinders of steam engines. An appeal to our readers brought out numerous statements concerning individual practice, eighty cases of which, as bearing upon the practical side of this discussion, are here tabulated so as to compare the amounts and costs per horsepower-hour and per million square feet of cylinder surface passed over.

2 A wide variation in these results, affected as they are by the design of the engines, the nature and finish of the surfaces, the quality, pressure and temperature of the steam, the adaptability of the oil to the service, method of application, and above all, degree of care and intelligence exercised in its use, would naturally be looked for, and it is there. If we compare the amounts of oil used per million square feet of surface wiped over by the piston, we find one man using 0.07 of a pint and another 5.94 pints, or 85 times as much, to lubricate the same area.

3 The higher value here mentioned, 5.94 pints, is, however, ex-

ceptional, the next highest value being 2.59 pints, and in the eighty cases analyzed only four go above 2 pints. In the high case the steam was very wet, which indicates that there are advantages to be gained by dry-steaming boilers, steam-pipe coverings and separators, besides the saving of heat units and cylinder heads. The low value, 0.07, is also very far from the average, which is 0.875, only eight of the cases being below 0.25.

4 The monetary importance of this question of cylinder lubrication is indicated by the column of costs per horsepower-hour, the average of which is 0.0075 of a cent. On a 1000-h.p. plant running 3000 hours per year, this would amount to $225 per year, and on a continuously running plant to between two and three times as much. For the high value, with wet steam, it costs 0.07 of a cent per h.p.-hr., which would be $2100 per 1000 horsepower, and a 3000-hr. year; while in the lowest case, at 0.00047 of a cent per h.p.-hr. the cost would be $14.10, a difference which would more than pay the wages of the average engine attendent in this one item alone.

5 It goes without saying that a very small part of the oil ordinarily used would lubricate the surfaces if it could be minutely and specifically applied to the surfaces in moving contact. Water is an effective preventative of such application. Oil will not attach itself to a wetted surface any more than water will adhere to an oily one.

6 The practical men to whom I talk are divided as to the best means of application. Some want the oil atomized, arguing that if it is diffused throughout the steam it will be carried to all the surfaces. Others prefer to keep the oil together and let it trickle down the side of the pipe and be washed along over the surfaces by the flow of steam, arguing that it is the valve and cylinder surfaces and not the exhaust pipe that they want to lubricate. Some prefer to use a force pump with a multiplicity of feeds, leading the oil positively to the various points of application. Some go so far as to drill the seats of Corliss valves longitudinally with several openings to the under surface of the valve; others, admitting the positiveness and efficiency of such systems, claim that they can get their oil where they want it with a simple sight-feed cup in the steam main, with fewer joints to leak, passages to clog and bright work to polish. The idea is prevalent among them that there is apt to be less trouble from a moderately smooth cylinder, one which shows the tool marks, than a highly polished one, the minute rugosities of the less highly finished surface aiding in the retention of the oil. This may have a bearing upon the fact often mentioned, that cylinders that have been run successfully without oil, as

LOW DISCUSSION.

No.	Estimated Horse-power	Descri	mt. per nated h. p. Pints	Cost per Estimated 1000 h. p. Hr., Cents	Amt. per 1,000,000 sq. ft. Rubbed over per Hr., Pints
1	1350	Corliss tripl	511	1.79	0.636
2	675	Cross com	69	6.46	1.76
3	675		99	4.33	1.25
4	975	Cross comp.	107	0.67	0.147
5	975	Corliss comp	639	2.23	0.89
6	650	Hamilton C	323	2.42	0.33
7	650	Hamilton C	392	....	0.40
8	575	Corliss comp	81	2.84	0.84
9	875	Corliss tand	229	1.14	0.37
10	450	Russell cross	555	4.16	0.58
11	650	Cross comp.	171	....	0.26
12	290	Cross com	22	15.80	4.42
13	290		11	8.32	1.17
14	200	Harris stand	92	6.90	1.22
15	200	Harris stand	65	....	0.50
16	400	Cross comp.	225	1.41	0.23
17	450	Corliss cross	927	6.95	1.10
18	290	Phoenix Iron	58	3.10	0.44
19	200	Russell tand	835	6.25	0.45
20	225	Corliss cro	86	21.45	5.10
21	225		43	10.72	1.36
22	225	Buckeye tan	111	8.33	0.69
23	450	Ingersoll cro	371	2.78	0.55
24	200	American Ba	65	....	0.98
25	180	Westinghous	461	4.32	0.32
26	160	Cross comp.	075	0.47	0.07
27	160	Tandem com	231	1.44	0.21
28	70	Ingersoll cro	185	8.89	0.88
29	45	Piston valve	927	2.90	1.16
30	450	Double eccen	111	3.47	1.57
31	290	Corliss.......	314	....	0.29
32	290	Harris Corlis	314	....	0.30
33	290	....	272	9.55	1.50
34	200	Greene......	455	....	0.39
35	160	Sioux Corliss	781	5.86	0.60
36	290	Harris Corlis	655	....	0.97
37	200	Corliss.......	835	2.61	0.87
38	130	New Brown..	908	6.81	0.63
39	250	Atlas single v	600	2.1	0.82
40	160	Ames........	375	6.00	1.27
41	160	Corliss.......	625	7.04	0.59
42	130	High speed a	115	0.72	0.089
43	130	High speed a	169	1.06	0.13
44	150	Piston valve	2	....	0.19
45	130	Slide valve th	769	3.36	0.63
46	120	Single valve a	15	4.02	0.88
47	160	Atlas four val	587	....	0.60
48	130	Harris Corliss	153	8.64	0.99
49	130	....	153	6.20	1.02
50	100	....	45	....	0.31
51	240	....	437	....	0.75
52	160	Wright Corlis	781	3.41	0.90
53	130	Nordberg Cor	538	14.38	1.47
54	100	Robt. Armstr	5	11.22	1.12
55	130	Corliss........	061	3.71	1.04
56	100	Corliss........	6	4.50	0.45
57	200	Slide valve...	09	....	0.14
58	160	Greene.......	187	1.46	0.25
59	100	Variable speed	45	2.82	0.38
60	85	Atlas automat	788	5.90	0.57
61	110	Ames	0	4.37	0.94
62	100	Automatic pis	25	7.8	1.09
63	130	Nordberg Corl	885	....	1.01
64	70	Ideal.........	186	5.18	0.73
65	70	Buckeye.......	428	....	0.92
66	100	Corliss........	49	....	0.47
67	70	Atlas single va	57	30.0	5.94
68	70	Corliss.........	19	....	0.82
69	70	Bates Corliss..	857	12.48	2.01
70	70	St. Louis Corli	643	4.01	0.47
71	70	New Brown...	428	13.38	1.04
72	50	....	1	4.81	0.58
73	100	Atlas..........	0	8.74	2.10
74	50	McEwen.......	8	7.50	0.42
75	70	Fitchburg.....	786	4.42	0.58
76	70	Slide valve....	185	....	0.90
77	40	Ball...........	175	....	1.97
78	50	Atlas..........	9	6.75	0.60
79	50	Center crank..	66	....	1.24
80	85	....	47	11.02	2.05
81	50	Slide-valve....	82	....	2.90
			014	6.19	0.97

TABLE 1 CYLINDER LUBRICATION. LUBRICATION AND LUBRICANTS. LOW DISCUSSION.

No.	Estimated Horse-power	Description of Engine	Cylinders	Stroke	R. P. M.	Steam Pressure	Sq. Ft. Rubbed over by Piston per Hour	Oil Used: Name or Description	Oil Used: Price per Gal. Cents	Total Amount Used per Hr., Pints	Cost per Hour Cents	Amt. Used per Estimated 1000 h. p. hr. Pints	Cost per Estimated 1000 h. p. Hr., Cents	Amt. per 1,000,000 sq. ft. Rubbed over per Hr., Pints
1	1350	Corliss triple expansion	20-34-52	60	65	...	1,084,000	...	28	0.69	2.42	0.511	1.79	0.636
2	675	Cross compound	h.p. 26	48	81	...	265,800	No. 725 cyl. comp.	75	0.466	4.37	0.69	6.46	1.76
3	675		l.p. 52	48	81	...	531,600	Capitol oil	35	0.666	2.91	0.99	4.33	1.25
4	975	Cross comp. St. Louis Corliss	22-44	48	85	140	706,500	No. 650 dark valve cyl.	50	0.104	0.65	0.107	0.67	0.147
5	975	Corliss compound	24-44	60	65	...	694,500	...	28	0.621	2.17	0.639	2.23	0.89
6	650	Hamilton Corliss cross comp	20-36	44	100	140	644,400	"600 W"	60	0.21	1.57	0.323	2.42	0.33
7	650	Hamilton Corliss cross comp	20-36	44	100	140-100 sup.	644,400	Oil of beeswax; 600 fire test Cyl. stock & acidless tallow	90	0.255	2.87	0.392	...	0.40
8	575	Corliss compound	18-34	48	85	...	555,900	...	28	0.465	1.63	0.81	2.84	0.84
9	875	Corliss tandem compound	24-42	48	65	110	539,600	Heavy body comp.	40	0.2	1.00	0.229	1.14	0.37
10	450	Russell cross compound	16-30	24	150	...	433,300	"600 W"	60	0.25	1.87	0.555	4.16	0.58
11	650	Cross comp. condensing	18-36	42	72	...	428,100	Improved high pressure cyl.	75	0.111	...	0.171	...	0.26
12	290	Cross compound	h.p. 18	36	86	120	146,000	Best grade cyl.	60	0.645	4.34	2.22	15.80	4.42
13	290		l.p. 34	36	86	120	276,000	Best grade cyl.	60	0.323	2.42	1.11	8.32	1.17
14	200	Harris standard cross comp	h.p. 16	18	200	110	150,800	Harris H. P. valve	60	0.184	1.38	0.92	6.90	1.22
15	200	Harris standard cross compound	l.p. 28	18	200	110	263,900	Rarus	...	0.131	...	0.65	...	0.50
16	400	Cross comp. St. Louis Corliss	14-28	42	85	...	393,000	No. 650 dark valve	50	0.09	0.56	0.225	1.41	0.23
17	450	Corliss cross comp	18-30	36	84	...	380,000	"600 W"	60	0.417	3.13	0.927	6.95	1.10
18	290	Phoenix Iron Works	14-24	18	210	130	376,100	...	43	0.167	0.90	0.58	3.10	0.44
19	200	Russell tandem compound	13-20½	20	210	...	368,200	"600 W"	60	0.167	1.25	0.835	6.25	0.45
20	225	Corliss cross compound	h.p. 16	36	84	...	126,800	Best grade cyl.	60	0.645	4.34	2.86	21.45	5.10
21	225		l.p. 30	36	84	...	237,500	Best grade cyl.	60	0.323	2.42	1.43	10.72	1.36
22	225	Buckeye tandem comp	12-21	21	200	...	363,100	"600 W"	60	0.25	1.87	1.111	8.33	0.69
23	450	Ingersoll cross compound	18-30	24	100	...	302,000	"600 W"	60	0.167	1.25	0.371	2.78	0.55
24	200	American Ball compound	10½-20½	12	285	...	277,600	...	...	0.273	...	13.65	...	0.98
25	180	Westinghouse automatic comp.	11-19	11	300	...	259,300	No. 725 cyl. comp.	75	0.083	0.78	0.461	4.32	0.32
26	160	Cross comp. Meyer valves	11-18	14	165	115	175,400	Best high grade mineral	50	0.012	0.075	0.075	0.47	0.07
27	160	Tandem comp. piston valve	10⅜-18	18	115	115	172,100	High grade mineral	50	0.037	0.23	0.231	1.44	0.21
28	70	Ingersoll cross compound	8-12	12	150	...	94,300	"600 W"	60	0.083	0.62	0.185	8.89	0.88
29	45	Piston valve	30	48	95	150	357,800	"Cyl. oil No. 10"	25	0.417	1.30	0.927	2.90	1.16
30	450	Double eccentric Corliss	30	48	84	150	318,000	"Cyl. oil No. 10"	25	0.5	1.56	1.111	3.47	1.57
31	290	Corliss	24	48	102	...	308,100	...	...	0.091	...	0.314	...	0.29
32	290	Harris Corliss	24	48	100	...	302,000	...	...	0.091	...	0.314	...	0.30
33	290	...	24	42	93	120	245,500	...	60	0.369	2.77	1.272	9.55	1.50
34	200	Greene	20	44	100	...	230,200	...	...	0.91	...	0.455	...	0.39
35	160	Sioux Corliss	18	42	105	...	207,900	"600 W"	60	0.125	0.94	0.781	5.86	0.60
36	290	Harris Corliss	24	48	65	...	196,200	...	...	0.19	...	0.655	...	0.97
37	200	Corliss	20	36	102	150	192,200	"Cyl. oil No. 10"	25	0.167	0.52	0.835	2.61	0.87
38	130	New Brown	16	36	125	110	188,800	Harris high-press. valve	60	0.118	0.88	0.908	6.81	0.63
39	250	Atlas single valve automatic	22¼	30	105	110	183,500	Buckeye cyl.	28	0.15	0.525	0.600	2.1	0.82
40	160	Ames	18	16	230	...	173,700	Capitol oil	35	0.22	0.96	1.375	6.00	1.27
41	160	Corliss	18	36	100	...	169,800	...	90	0.1	1.12	0.625	7.04	0.59
42	130	High speed automatic	16	16	250	115	167,800	High grade mineral	50	0.015	0.094	0.115	0.72	0.089
43	130	High speed automatic	16	16	250	115	167,800	High grade mineral	50	0.022	0.14	0.169	1.06	0.13
44	150	Piston valve engine	17½	16	220	80	161,500	Light colored oil	...	0.03	...	0.2	...	0.19
45	130	Slide valve throttling governor	16	30	126	95	158,200	Capitol	35	0.1	0.44	0.769	3.36	0.63
46	120	Single valve auto. Fitchburg	15½	14	275	...	156,200	...	28	0.138	0.48	1.15	4.02	0.88
47	160	Atlas four valve	18	15	220	125	155,900	Rarus cyl.	...	0.094	...	0.587	...	0.60
48	130	Harris Corliss	16	36	100	65	150,900	Harris std. grade	60	0.15	1.12	1.153	8.64	0.99
49	130	...	16	38	92	...	146,600	Compounded oil	43	0.15	0.81	1.153	6.20	1.02
50	100	...	14	14	280	...	143,200	Model cyl.	...	0.045	...	0.45	...	0.31
51	240	...	22	23	106	...	140,500	"Eureka cyl."	...	0.105	...	0.437	...	0.75
52	160	Wright Corliss	18	42	70	85	138,800	Capitol	35	0.125	0.55	0.781	3.41	0.90
53	130	Nordberg Corliss	16	36	90	...	135,800	Valvaline	75	0.2	1.87	1.538	14.38	1.47
54	100	Robt. Armstrong automatic	14	14	260	125	133,500	"600 W"	60	0.15	1.12	1.5	11.22	1.12
55	130	Corliss	16	42	75	...	132,100	...	28	0.138	0.48	1.061	3.71	1.04
56	100	Corliss	14	36	100	...	132,000	"600 W"	60	0.06	0.45	0.6	4.50	0.45
57	200	Slide valve	20	30	80	80	125,800	...	...	0.018	...	0.09	...	0.14
58	160	Greene	18	30	85	...	120,200	"600 W"	60	0.03	0.22	0.187	1.46	0.25
59	100	Variable speed, St. Louis Corliss	14	36	90	135-140	118,800	No. 650 dark valve cyl.	50	0.045	0.28	0.45	2.82	0.38
60	85	Atlas automatic	13	18	190	125	116,800	"600 W" vacuum cyl.	60	0.067	0.50	0.788	5 90	0.57
61	110	Ames	15	14	212	...	116,600	Capitol	35	0.11	0.48	1.0	4.37	0.94
62	100	Automatic piston valve	14	14	223	...	114,500	High grade heavy	50	0.125	0.78	1.25	7.8	1.09
63	130	Nordberg Corliss	16	32	85	110	113,800	Dark heavy cyl with graph.	...	0.115	...	0.885	...	1.01
64	70	Ideal	12	12	200	100	113,100	Capitol cyl.	35	0.083	0.36	1.186	5.18	0.73
65	70	Buckeye	12	21	165	...	108,900	...	...	0.1	...	1.428	...	0.92
66	100	Corliss	14	36	80	...	105,800	...	...	0.049	...	0.49	...	0.47
67	70	Atlas single valve automatic	12	18	180	110 very wet	101,800	Buckeye cyl.	28	0.6	2.1	8.57	30.0	5.94
68	70	Corliss	12	36	90	...	101,800	...	...	0.0835	...	1.19	...	0.82
69	70	Bates Corliss	12	36	88	100	99,500	Capitol cyl.	35	0.2	0.87	2.857	12.48	2.01
70	70	St. Louis Corliss	12	36	85	135-140	96,350	No. 650 dark valve cyl.	50	0.045	0.28	0.643	4.01	0.47
71	70	New Brown	12	34	90	80	96,200	Franklin oil	75	0.1	0.94	1.428	13.38	1.04
72	50	...	10	14	260	80	95,400	Capitol cyl.	35	0.055	0.24	1.1	4.81	0.58
73	100	Atlas	14	20	130	80	95,400	Capitol cyl.	35	0.2	0.87	2.0	8.74	2.10
74	50	McEwen	10	12	300	...	94,400	W. P. Miller's cyl. comp.	75	0.04	0.37	0.8	7.50	0.42
75	70	Fitchburg	12	30	100	90	94,400	...	45	0.055	0.31	0.786	4.42	0.58
76	70	Slide valve	12	14	210	150	92,500	Flake graph. with eng. oil	...	0.083	...	1.185	...	0.90
77	40	Ball	9	12	300	...	84,750	Pomo oil	...	0.167	...	4.175	...	1.97
78	50	Atlas	10	12	240		75,400	...	60	0.045	0.34	0.9	6.75	0.60
79	50	Center crank	10	15	170	...	66,800	...	...	0.083	...	1.66	...	1.24
80	85	...	13	17	105	100	60,900	"600 W"	60	0.125	0.94	1.47	11.02	2.05
81	50	Slide-valve	10	12	250	100	31,410	Premium valve oil	...	0.091	...	1.82	...	2.90
		Averages										1.014	6.19	0.97

is common in marine pratice, and which are then lubricated, refuse to run quietly thereafter without oil and plenty of it.

7 There is no doubt that much of the expense of cylinder lubrication and much of the annoyance, interruption of service and cost of repairs due to ineffective cylinder lubrication, can be avoided by a careful selection of the oil for a particular service. It stands to reason that a heavy low-pressure piston and the massive valves of a low-pressure cylinder at a comparatively low temperature, when using the moisture-charged steam from the high-pressure cylinder, will call for a different lubricant than will the lighter but hotter parts of the initial stage.

8 For the lubrication of cylinders using superheated steam and in internal combustion engines an oil of high viscosity and flash point is required. Some of our readers are puzzled to know how such cylinders are lubricated by oils which vaporize at temperatures less than those of the working fluid. The flash points are, however, taken under atmospheric pressure, and it is evident that the oil will remain liquid and sufficiently viscous under the high pressures used, and especially upon the surfaces of the cylinder which do not attain the temperature of the gas or steam, while it would become too limpid to serve as a lubricant or would even vaporize at atmospheric pressure and the temperature of the working fluid.

9 Attempts are made to improve the effectiveness of the lubrication of gas-engine cylinders by timing the injection of the oil. It is to be expected that a given quantity of oil will go further when injected into the piston grooves on the exhaust stroke, where it has that and the succeeding induction stroke to spread over the cylinder walls before ignition, than if injected directly into the cylinder when combustion is going on, in which case it would be only an expensive fuel. The delicacy of such timing may be appreciated when it is recalled that the speed of the piston at the center of the cylinder, where the oil is ordinarily introduced, is about 1200 ft. per min., at which rate it would take a piston one foot in thickness one-twentieth of a second to pass the oil hole, not a long time to get a column of oil into motion and stop it again. I believe sone builders are introducing the oil at the ends of the stroke while the piston is dwelling on the center. It would be interesting if some of the gas-engine men would tell us of present pratice with regard to location of feed, time of injection, and what kind of lubricant gives the best results.

Dr. D. S. Jacobus. At one time at the Stevens Institute of Technology a great many tests were made on lubricating oils. Later on

when anyone sent an oil to be tested we wrote that our experience in making friction tests had convinced us that results obtained in a laboratory gave no indication of the value of an oil as a lubricant in practical service; and further, that if we made friction tests a statement as to their lack of reliability would be incorporated in the report. As a rule those wishing to have friction tests made did not then send oils to us.

2 The above decision was not reached until a great many tests had been made, with various forms of machines. Professor Denton conducted most of these tests, and it was my privilege to be associated with him in the work. In one machine a car axle was mounted on roller bearings and so arranged that the full load that would ordinarily come on the journal could be applied and measured. This machine was run at speeds corresponding to railroad practice, all the conditions of service being copied as closely as possible. The journal was given an end play and a blower was used, to blow air over the entire journal at the speed at which a car would ordinarily travel. Very accurate results were obtained with this machine, which was run for eight or ten hours a day for a number of months.

3 Tests were also made with another machine for determining the friction inside a steam engine cylinder. The frictional force exerted on the piston rings was measured in a way that eliminated the effect of inertia. Before the advent of the above machines we had spent many hours with a Thurston friction oil-testing machine in which the pressure is applied in two directions on the journal at all times, which is not the usual case in practice; and the idea in constructing the special machine was to place the oils under as near service conditions as possible.

4 One feature which was very definitely shown was, that if friction tests are to mean anything at all, the friction of one oil should be compared with the friction of another. The condition of the journal greatly affects the amount of friction, and if a journal is worked down to give the best results, it requires a long time to get it back into shape, should it accidentally become abraided. In fact, as the brasses wear away, the variation in form of the bearing surface will sometimes change the coefficient, and it is no exaggeration to say that with a given oil, results varying by 100 or 200 per cent will be given, by a journal in the same condition as far as the eye can see.

5 In comparing the results with a standard oil, we first determined the friction of the standard oil, then the friction of the oil in question, and then went back to the standard oil. If the two tests with the standard oil agreed substantially with each other we reported the per-

centages of friction, but if they did not agree we continued making experiments, first with one oil and then with the other, until there was an agreement.

6 We made a number of attempts to measure the wearing qualities, or durability, of an oil and finally concluded that there is no such thing as wearing out an oil. Any test of this sort that can be gotten up is simply a measurement of how the oil sticks to the journal and lubricates it.

7 Professor Denton eventually developed the idea of tracing out the lubricating qualities of an oil by observing how it acted in the practical field. One young man connected with the department made a number of trips across the ocean on freight steamers, to determine the quality of one oil as compared with another for marine service. Tests were also made on locomotives and other classes of engines. To me was assigned the task of determining which of two oils would be the best for service on transatlantic liners, by making a trip from New York to Liverpool on the American Line steamer New York, and I can assure you that there is much besides friction to be considered in selecting the oil which will be the best for that sort of work. I do not blame engineers for refusing to try a substitute when they obtain an oil that suits them. There is enough strain in keeping up a maximum speed in an ocean liner without having to worry about the quality of the oil. As far as I know, oil of the composition decided on is still used for the work.

Chas. A. Hague. In lubricating steam cylinders, the condition of the steam makes a difference as to how the oil will hold to the surface. I know of cases in which a very good cylinder oil, recognized for many years, failed entirely to lubricate the valves and cylinders of a large Corliss engine. I think it cost $1.10 a gallon, and after a great deal of experimenting it was found that an oil which cost 30 cents a gallon and was inferior in appearance, lubricated the cylinder perfectly. Apparently the steam was so wet, and the better oil so viscous, that the valves and cylinders did not get any lubrication at all, while the cheaper oil was thinner and more easily distributed.

George A. Orrok. Mr. Souther's remarks concerning brass bearings and babbitted bearings have reminded me of my own experience with bearings, not in automobile engines, but in the larger steam engines. Formerly, it was customary to make the cross-head box and the crank-box of brass, sometimes babbitted, but very rarely.

The boxes were lubricated as well as possible, but they would become hot.

2 We finally discontinued using brass, and I think every large engine builder today uses cast-iron or steel boxes and lines them with babbitt. The trouble from hot bearings has disappeared. I do not think this is due to the use of a larger amount or a better quality of oil, but to the superior construction of the boxes themselves.

Prof. P. E. Walter.[1] It is well recognized that the usual tests for viscosity, flash temperature and acidity, are valuable for their discriminative testimony, but every engineer is looking for some conclusive test which will give positive information without entailing the expense, time and frequent loss, occasioned by a long-time trial in actual service.

2 One phase of the flash temperature which is easily overlooked, is its relation to the rate of loss by evaporation. Practically all our commercial and lubricating oils are more or less complex mixtures of hydrocarbon compounds, any given oil being made up of compounds vaporizing at different temperatures. The lightest hydrocarbon in the group fixes the flash temperature, and also influences in large measure the evaporation loss. This loss is often a serious consideration, since in continuous oiling systems the gradual thickening and deterioration serve to increase the difficulty of handling the oil, to increase the friction loss in the bearings and cause an actual loss of oil. It seems natural to expect that some relationship should exist between flash temperature and loss by evaporation.

3 With a view to such relationship, the writer carried out in his laboratory a test for evaporation loss under several different temperatures, of three oils whose flash temperatures were carefully determined. While the data thus found are insufficient to establish fully a law of relationship, the results may be expressed with a fair degree of approximation by an equation of the form

$$L = C\left(\frac{t}{100}\right)^3 (400-T)^2$$

where L = loss in per cent of weight in 300 hours.

t = temperature of oil in use.

T = flash temperature.

For oils flashing between 325 deg. and 375 deg. fahr., C may be taken at a value of 0.0003. For oils flashing below 300 deg., C is about

[1] University of Kansas, Lawrence, Kans.

0.00075. These figures apply to oils at temperatures above 100 deg. fahr., to which oil confined in closed crank cases of engines and in some hollow bearings may be subjected for long intervals. The great difference in evaporation loss between oils flashing below 300 deg. and those flashing only 30 or 40 deg. higher, is due to the presence in the former of light hydrocarbon compounds, left there by an inferior method of refining. Such low-flash test oil can of course, be seriously considered only for intermittent service in cool places.

4 The viscosity test is a valuable one, but cannot be taken as a direct measure of the lubricating value of an oil. My own experiments, with several oils all of which are capable of bearing a given load and otherwise performing satisfactorily in service, have shown that the one having the least viscosity will give the lowest coefficient of friction. To the careful practicing engineer who is looking for the best lubricant for his work, and who has found out by the test of actual service that certain oils will operate successfully in his plant, this gives a method of making a final selection.

5 Care must be exercised in following out this method, however, since an oil might be under too great a bearing pressure and still be able to keep the bearing temperature down to a permissible point. Some form of endurance test should be made to yield information on this point. Furthermore, in making the viscosity test, each oil should be tested at several temperatures over a range including that at which the oil is used, remembering that the actual oil film in the bearing is probably several degrees warmer than can possibly be registered by any thermometer set in the metal of the bearing.

6 By plotting curves of viscosity for varying thermometers it is seen that oils with viscosities markedly different at some ordinary temperature such as 60 deg. fahr., may be virtually identical at the temperatures at which they will run in a given bearing, or even reversed in relative values. Tables of viscosity-values of oils at a common temperature are virtually worthless. As a complementary fact it may be noted, that a bearing running moderately warm is not of necessity a cause for condemning one oil in favor of a second. It may simply mean that a condition of temperature equilibrium is reached with the first at a different point from the second, and a consultation of viscosity records at the proper temperatures is necessary before a decision can be intelligently made.

7 It would be both convenient and interesting if data could be obtained by which one could calculate in advance, by some exact method based on a knowledge of the oil, the amount of work lost in

friction of lubricated bearings for different conditions. This would require the determination of the coefficient of friction on special friction machines, and the experience of the writer with three different types leads him to the conclusion that this is impossible. Conditions in practice cannot be duplicated on a special machine; or at most but a limited number of sets of conditions could be so duplicated, and absolute values of the friction coefficient thus obtained are questionable.

8 The friction machine plays an important part in the investigation of a lubricant, however, provided the bearing is always supplied with oil in the same manner. This question of supply is an extremely important one, because a vital factor in the lubricating value of an oil is its ability to spread over the surface. In a special machine, an oil which would not spread well with the system of supply and distribution adopted would give poor results in comparison with some other oil, while under different conditions of supply better results might be obtained.

9 A laboratory investigation must conform to the injunction to vary one thing at a time. The writer is of the opinion that perfect lubrication is necessary in comparing the true lubricating values of different oils, the most reliable method being immersion of the bearing in an oil bath. With a limited supply of oil imperfect lubrication may exist, and the result is neither scientifically exact or of any practical value to a user whose conditions are in any way different from those existing on the machine. There are no laws of imperfectly lubricated surfaces. These considerations bring a cross fire on the man who is striving to reduce both friction and oil bills, making it necessary for him to study the problem of application as well as the oils themselves.

10 In a perfectly lubricated bearing, the resistance to motion is due entirely to fluid friction of the lubricant. This is true whether the motion of the fluid elements be considered as that of one extremely thin layer sliding over and under its adjoining thin layers, below and above, or that of round particles or balls rolling between the bearing and the journal. In either case there is shearing action between the particles, due to the motion. Viscosity is the indication of the magnitude of this shearing force, and hence for perfectly lubricated bearings low viscosity is an index of a good lubricant. But it is plain that to be a good lubricant the oil must first possess the power to form films and maintain those films against pressure, which power is indicated by what is termed "surface tension" of the liquid.

11 The thickness of the film has a double significance, also, since the thicker the film the greater may be the irregularities in the surfaces without an actual puncture; and the thicker film gives a lower rate of motion of one sliding layer of oil over its neighboring layer, if we adopt the sliding theory of lubrication, and consequently a lesser force of resistance.

12 May we not expect then, that the two tests for strength and thickness of oil films would help in the selection of lubricants? Such tests might, in some measure, be applied to greases as well as oils. In the opinion of the writer, the reason why viscosity is not a positive indication of the lubricating value of an oil is simply that it tells but one portion of the story, while the remaining portion must be told by an investigation of oil films. Such an investigation should answer the following questions:

a Do thick films give less friction than thin films?
b Does high surface tension indicate a power to resist high pressures?
c Does high surface tension indicate a power of the oil to spread well and so give perfect lubrication?

Standard apparatus for making the determinations, and a system of numerical relationships, should also be worked out.

MALCOLM MCNAUGHTON.[1] The paper is most interesting, yet lacks one important detail of information, needful if we are to draw any conclusions from the tests. This omission is a test of oils, without graphite, of such character as would indicate their entire suitability to the stated conditions of velocity and pressure. The velocity (115 ft. per min.) and pressure (150 lb. per in.) would indicate the use of an oil of quite high viscosity. The tests did not include any such oils, excepting the cylinder oils, and these were tried at such a temperature that their viscosity was low. Because of this omission we are not able to make any comparison between the best possible lubrication by oil alone, and by oil or water mixed with graphite.

2 The effect of graphite in lubrication may be best considered if we keep in mind that the total friction of a lubricated journal is the sum of the internal friction of the lubricant itself, plus the friction due to the intermittent or continuous contact of the metallic surfaces. It is plain that ordinarily as one increases the other decreases, though

[1] Joseph Dixon Crucible Company

of course we can conceive of cases where a change in lubricant might be followed by an increase or decrease of both components. We may never be able to determine in any particular case what proportion of the total resistance is due to the viscosity of the lubricant and what to metallic contact; nor is this necessary if we know that the use of any particular oil is giving us the best obtainable economy.

3 The addition of graphite, such as was used by Professor Mabery in his tests, to an oil of 154 rated viscosity, even up to 1 per cent by weight, caused no appreciable change in viscosity. A pipette of 100 cubic centimeter capacity allowed the oil, with and without graphite, to flow through in exactly 151 seconds in every trial. Therefore if there is no change in the viscosity of the oil due to graphite, any reduction of total friction must be found in the reduction of the friction between the metallic surfaces. That interposing of graphite between frictional surfaces reduces the friction due to metallic contact is so well known as to need no demonstration.

4 From the preceding statements it naturally follows that where the metallic frictional component is relatively large the graphite effect will be large, and on the other hand where it is small the effect of the graphite will be relatively small.

5 The light oils selected by Professor Mabery for his tests were well suited to show the action of graphite in reducing metallic friction though we are left in the dark as to the effect if oils of suitable viscosity had been used. Keeping in mind what has been said, the lubricating value of a mixture of graphite and water is not remarkable; for as the graphite reduces the metallic friction, the ratio of the fluid friction to the total friction is greatly increased, and the total friction lessened. Many will take exception to the statement that water is "completely devoid of lubricating qualities." It lacks but one quality of the ideal lubricant, and that is ability to keep the metallic surfaces apart. When this defect is provided against by pressure, water gives most perfect results. The matter in Professor Mabery's paper does not give us any basis for forming an opinion as to the relative merits of lubrication by water and graphite, as compared with lubrication by oil alone; to make such a comparison we must know not only their relative efficiency as lubricants but their relative cost, in order to determine their relative economy. If the water plus graphite costs more per gallon than the oil alone, and lubricates no more efficiently, then of course it becomes less economical in use.

6 We may now turn to a consideration of the way in which

graphite serves to reduce journal friction. For purposes of illustration let us consider the case of a fast running journal and a well fitting bearing in which the projection of any irregularity from the normal surface is less than the dimensions of the particle of graphite. The film of oil has its greatest velocity at the surface of the revolving journal and least at the surface of the bearing. Any particle free to move in the oil will be thrown outward and finally come in contact with the surface of the bearing. The journal's motion is always slightly eccentric so that in time every part of the bearing surface has been in contact with the journal. These two forces serve in time to produce on the bearing a veneer of graphite more or less perfect. But in case the irregularities of the frictional surfaces are greater in depth than the dimensions of the graphite particle, it must be seen that the graphite veneer may not be formed at all or be so long delayed in formation as to allow serious trouble to develop. Even approximately perfect bearings are hard to secure and the more they depart from a perfect condition the greater the difficulty in getting the true graphite surface.

7 It is clear that the rougher and more irregular the surface of the bearing the greater the size of the particle should be, in order to prevent the high points from coming in contact with the journal. This fact appears to limit the use of this very finely divided graphite to bearings of the most perfect character and which are the least likely to give trouble. Professor Mabery calls particular attention to the importance of maintaining a high state of perfection in all bearings, which of course is entirely true. But in fact, the normal condition of the average bearing is quite otherwise, and this is the type of bearing which we have most to consider. The statement that the artificial graphite is more unctuous than the natural graphite is denied absolutely. All high grade graphite is unctuous, whether made by nature or artificially.

8 This characteristic is usually determined by rubbing between the finger tips. On trial it will be noticed that while good natural crystalline graphite appears unctuous without pressure, the finely ground artificial graphite must be rubbed with some pressure and brought to a polish before the same effect is secured. The smoothness of this surface does not compare with that of every particle of crystalline graphite, while it is much less resistant to wear.

T. C. Thomsen.[1] Professor Mabery quotes the opinon of well-known experimenters on oil-testing macnines, that "such experiments do not afford results comparable with those in actual practice." This is also the conclusion to which the writer has arrived after having carried out a great number of comparative friction tests on machinery of the most varied description.

2 As Archbutt suggests, it is the quality of "oiliness" or "greasiness" that is so very important when judging the lubricating value of an oil; and this characteristic, viz., the adhesiveness of an oil to different bearing metals, is a factor that can never be, or at least has not been up to now, brought under the control of the laboratory. Two oils of practically the same gravity, viscosity, flash point, fire point and color, and both pure mineral, but otherwise made from different crude oils or by different methods, may in actual practice show a difference in friction of as much as 14 per cent. This has been recently proved in Germany, the test being carried out by the electro-technical department of the Bayerische Landesgewerbe-Anstalt in a textile mill.

3 Professor Mabery, while admitting that the conditions on the friction testing machine should be as nearly as possible the same as the conditions under which the oil is actually to be used, has carried out his experiments on a machine which to my mind is as far away as possible from being representative of any bearings at present employed in any kind of machinery. The diameter of the testing machine shaft is 1 in., and the length of the journal about 12 in. In order to keep this bearing and journal in satisfactory operation Professor Mabery found it necessary, not only to mill the journal and bearing to mechanically true surfaces, but by repeated careful milling an even higher degree of permanent evenness had to be maintained. Also, bronze was found unsuitable as a bearing metal and white metal was adopted. (The explanation of this is no doubt that for the abnormal length of the bearing in proportion to the diameter, white metal being more yielding than bronze produced a more uniform bearing pressure.) Further, means were provided for examining the surface of the journal, which means that the oil film was broken, making it difficult for the oil to perform its function, namely that of forming a separating film between the journal and the bearing. If finally the sides of the brasses were not eased away (chamfered), this would form another condition of the bearing which is not, or should not be, met in good engineering practice.

[1] Address, care Wm. F. Parish, Jr., Deutsche Vacuum Oil Company, Kaiser Wilhelmring 4, Cologne, Germany.

4 After Professor Mabery has succeeded in making his most exceptional bearing work satisfactorily, he concludes that the same amount of care should be applied in ordinary factory operations. This suggestion is, to say the least, absurd, as it would increase the cost of all machinery several times and it would not be possible to maintain such a standard of accuracy for any length of time.

5 The net result of Professor Mabery's tests is, that when graphite has been used and the supply of lubricant cut off, it will take a longer time for journal and bearing to "seize" than if no graphite had been employed. This is easy to understand, and has been known ever since graphite was first employed as a "lubricant."

6 Wherever journals or bearings, or say the metal of steam-engine cylinders, are of a porous nature, graphite will fill the pores, acting as a leveller of the surface. If a bearing is cracked and it is desired to continue operation, as otherwise a decrease in the works output might result, graphite can be used to fill the crack, preventing the oil from escaping and thus making it possible to keep the machinery operating the desired length of time.

7 As to the actual friction of bearings in good condition, whether graphite be used or not the friction is practically the same, as shown by Professor Mabery's experiments.

8 Referring to Fig. 2, the coefficients of friction with Nos. 2, 3 and 4 are practically the same. That it is possible to use fuel oil, kerosene and water will be explained by the low bearing pressure and exceedingly good finish of the journal and bearing.

9 It will be understood that a bearing pressure of 70 lb. per sq. in. on this particular bearing, due to its better finish, can be more easily sustained than the same pressure in an ordinary bearing. In other words, if the same mixtures of graphite, fuel oil, kerosene oil or water, were used on ordinary bearings, they might not be able to sustain more than 15 to 20 lb. per sq. in. It will be noticed that Fig. 3 does not contain any curve for water and graphite alone, probably because even with this low pressure it was not possible to operate the bearing without trouble.

10 Referring to Fig. 4, this shows that with an oil consumption of 6 drops per min. it was not possible to maintain an unbroken film. This shows either that the oil must have been of poor quality or that the edges of the bearing were not chamfered, as being sharp they would scrape off the oil film.

11 The differences in the coefficient of friction as shown by the different curves, Nos. 2, 3 and 4, are not greater than might be antici-

pated on any friction machine, and do not prove that the admixture of graphite reduces the coefficient of friction. What the curves do show is that with graphite, after the oil supply has been cut off, it takes a longer time for the friction to increase rapidly than when oil alone is used.

12 As to Figs. 6, 7 and 8, the conditions under which these cylinder oils are used in actual practice, with the steam engine piston moving to and fro over a film of oil mixed with water from the steam, and the oil more or less emulsifying with the water and being heated to a temperature much higher than 210 deg. fahr., are so different from the conditions in the testing machine, that for the purpose of comparing the lubricating qualities of the different oils they have no value. Further the bearing pressure in steam engines between the piston rings and the cylinder walls should always be very slight, nothing like 1200 lb. per sq. in. The temperature of the experiments is stated as 210 deg. fahr., but what is most important to know (from Professor Mabery's point of view) is the actual temperature of the oil film. This I take it, it has not been possible to obtain, and a slight difference in temperature of the oil film would make a very considerable difference in the viscosity of the oils (being cylinder oils), which again would have a considerable influence on the coefficient of friction.

13 Judging from Figs. 7 and 8, it looks as if the trials with the straight oils (600W and Galena cylinder oils) had not been carried out under the same temperature of the oil film as the trials with the oil mixed with graphite. For instance, in Fig. 7, if the straight oil had been a little higher in temperature the coefficient of friction for the first portion of the diagram, up to the point of oil cut-off, would have been lower, and after that point, the oil being warmer, it would be squeezed out of the bearing more quickly, with a resultant sharper rise in the curve for the coefficient of friction, thus getting closer to the curve for the graphite and oil mixture. The same remarks apply to Fig. 8. This suggests that the coefficient of friction, if the temperature of the oil film had been the same in every case, probably would have given the same diagram for the oils used straight as for the oils mixed with graphite.

14 In conclusion, as to the practical question of using oils mixed with graphite, Professor Mabery says that it is possible for graphite in the deflocculated condition to distribute itself readily in water, but that it is quickly precipitated by impurities. As it is not possible for any practical purpose, except perhaps for certain special conditions, to apply a mixture of water and graphite, it will be necessary, in order

to carry out Professor Mabery's suggestion, to use a mixture of oil and graphite, and I presume that also in this case slight impurities would cause the graphite to be precipitated. Therefore if it were used in oil cups or lubricators, without mechanical continuous mixing, the graphite would precipitate and choke the oil channels leading to the bearings. Further, it is the usual practice in many engineering works, to filter large quantities of oil, for re-use, after being used on bearings. It is well known that waste oil mixed with graphite quickly clogs up the filter pads and makes them inefficient, necessitating frequent cleaning and recharging with filter material.

15 As to oil circulating systems, where a certain quantity of oil is continuously circulated through the bearings, returning to a filter and passing an oil pump, and again forced out to the bearings, in such a system also any admixture of graphite would in time mean accumulations of deposit in the lubricating pipes, eventually putting the system out of action if not frequently cleaned out.

The Author. The interesting scientific and practical observations accumulated in Dr. Conradson's extended experience in handling lubricants, and here placed on record, will be extremely useful as representing the present state of knowledge on this subject. His results from the testing of oils on the larger machine indicate what I have noted in a great variety of lubricants, that the coefficient of friction very materially diminishes with increasing pressures.

2 Dr. Conradson's tabulated data contain much practical information. Table 1 demonstrates the deterioration from long continued use, that every oil must undergo, by evaporation with consequent changes in congealing point and in viscosity, and lessened durability. His method of testing under constant temperature affords an accurate measurement of the total energy of friction. The total heat absorbed by the water used in cooling may readily be ascertained.

3 As Mr. Davis suggests, temperature tests on factory bearings may give valuable indications as to the comparative efficiency of oils. Why may not some form of friction indicator be set up in any factory for observations on friction?

4 It takes a good oil, as Mr. Souther explains, to stand the hard usage on automobile bearings, especially in crank-case lubrication; decomposition to a greater or less extent is sure, with deterioration in lubricating quality, even to carbonization at increased temperatures.

5 The data of comparative cost of lubrication, tabulated by Mr

Low, illustrate forcibly the great waste that may follow from careless selection of oils, but more especially from lack of attention to the proper use of lubricants.

6 The suggestion by Dr. Jacobus that a standard oil be selected for control of the condition of bearings, seems to be about the only way whereby these conditions may be made dependable for accurate work; in long-continued tests an operator can decide quite accurately from experience, with occasional use of this aid.

7 Mr. Orrok's experience with brass bearings appears to indicate as I have observed on the experimental scale, that babbitt adapts itself more readily to the condition of the journal, especially in heavy work, with less heating and less friction, perhaps by what may be termed metallic flow.

8 As Mr. Walter remarks, continued use of an oil, especially with considerable agitation, as in flooded bearings or in automobile-crank cases, causes evaporation of the constituents, with consequent thickening and gumming, it may be to carbonization with high temperatures. His experience with reference to viscosity coincides with my observations, that economic selection and use of an oil must depend on the intelligence of the engineer for proper appreciation of the data of chemical, physical, and frictional tests, and ability to adapt them to particular conditions. His lucid explanation of the manner in which the film operates, in relation to viscosity and surface tension, should, I think, include the important influence of oiliness. In answer to his question *a*, I would suggest that the thinner the film the less the internal viscosity, and the lower the coefficient of friction. I believe this holds true in a large number of observations, extending through the wide range of oils from kerosene to the heavier cylinder oils and greases.

9 To answer fully the several pertinent questions suggested by Mr. McNaughton would require more space than is available. The speed in these tests is not 115 ft. per min., as he infers, but the 450 revolutions are equivalent to about 400 ft. per min., and the viscosity of the great number of automobile oils examined during the past year is less than 300 sec. Saybolt at 70 deg. The viscosity of the particular oil shown in the chart is 196 sec. Saybolt, at 70 deg. I did not include the data collected on all these oils, as it would have extended the paper to an undesirable length. I may state, however, that every oil examined gave a higher coefficeint alone than when mixed with 0.35 per cent graphite; more than this amount is not desirable, as I explained.

10 I think Mr. McNaughton will find an answer to his question, as to the relative efficiency of water and graphite, as compared with oil, alone, in Fig. 2, which gives the coefficient of an automobile oil alone as nearly 0.02, and that of water and graphite as 0.01; for as mentioned above, in every oil examined graphite diminishes the coefficient.

11 As to the question of comparative cost, I think it is a fact that lubrication by water and graphite is considerably less expensive than lubrication with oils; certainly this is true for automobile lubricants that cost the consumer 50 cents to $1 per gal.

12 Referring to the question of the detailed action and movement of the graphite particles on the journal and bearing, it is a distinguishing quality of deflocculated graphite that it readily forms a continuous coherent film, and serves especially as a surface-evener, by filling up irregularities of the frictional surfaces, so that they approach approximately the condition of perfect bearings. I have recently seen striking examples of this action on journals and bearings run for many months continuously, with lubricants containing the regulation proportion of graphite. A much smaller amount of oil with graphite, and of the oil alone, is required to support the same load and speed after this long-continued use of the graphite. This seems to be an important function of the graphite.

13 I must take issue with Mr. McNaughton as to the comparative specific qualities of deflocculated graphite, which differentiate it from the other forms and adapt it especially for lubrication.

14 Referring to Mr. Thomsen's strictures on the Carpenter machine and my manipulation of it, I would remind him that I suggested the adaptation of a machine to the particular conditions of factory operation. It is not true, as he asserts, that the friction is the same whether oil alone or oil with graphite be used. In every oil I have tried, deflocculated graphite reduces the friction, as is shown on the charts accompanying the paper. The "net results" of my tests are, reduced friction, and longer life of a lubricant carrying deflocculated graphite. This systematic and dependable use of graphite has been known since it has been used in deflocculated form.

15 Figs. 6, 7 and 8 do show plainly that graphite reduces friction in heavy oils at steam temperatures, and that a film of graphite forms and supports the bearing for a long time after the oil is shut off. In Fig. 7 and Fig. 8 the temperatures of the oil films may have been slightly lower than that of the bearing, but not much; at any rate the

temperature of the oil alone, and that of the oil with graphite, were the same.

16 Mr. Thomsen's allusion to my statement that graphite is quickly precipitated from water by impurities leads me to explain that this statement was intended to apply to the deflocculation of the graphite by alkalies and acids, or in general by electrolytes. It does not apply to the ordinary impurities in lubrication either by water and graphite or by oils containing graphite.

17 Mr. Thomsen's statement, that "it is well known that waste oil mixed with graphite quickly clogs up the filter pads and makes them inefficient," shows that he has yet to learn that deflocculated graphite filters readily through any medium. But this use of the mixtures which I have tested, dependent on filtration, is not contemplated in their proper economic application.

CAST-IRON FITTINGS FOR SUPERHEATED STEAM

THREE PAPERS BY PROF. IRA N. HOLLIS, PROF. EDW. F. MILLER AND ARTHUR S. MANN, PUBLISHED IN THE JOURNAL FOR DECEMBER 1909.

ABSTRACT OF PAPERS

The paper by Professor Hollis stated that the examination of cast-iron fittings after long exposure to superheated steam considerably above 500 deg. in temperature has disclosed distortion, cracks and permanent change of shape. Test pieces taken from such fittings have shown irregularity of strength and apparently some reduction. The case has not been conclusively made out by exhaustive tests. The theories as to chemical change rest on doubtful grounds.

Three large cast-iron fittings, two of which had been exposed in service longer than one year, to superheated steam of 578 deg., were burst by hydraulic pressure and found to be amply strong for all purposes, although the test pieces subsequently cut from them failed fully ten per cent below what should have been the tensile strength of air-furnace gun iron out of which they were made. No tests were made of the original metal and nothing conclusive was proved except that the bursting strength of a 14-in. T was 1650 lb. per sq.in.

A calculation of the stresses set up by the expansion of the long line of pipe in which the T's were placed gives a possible stress of nearly 4000 lb., and even more, due to expansion exclusive of the stresses imposed by the steam pressure. It would seem therefore that the deterioration, if any existed, was due to the absence of expansion joints.

The paper by Professor Miller gave the results of tensile tests as specimens of cast iron, gun iron and steel, certain of which were first subjected to the action of superheated steam. While the tests were so few as not to justify many conclusions, it was evident that the metals had suffered a loss in strength due to their exposure to the steam.

The paper by Mr. Mann stated that there is a growing feeling among engineers that nothing but steel will answer as a container for highly superheated steam. A sound high-grade steel casting is practically unaffected by any reasonable superheat and if the user is willing to pay ten cents or twelve cents per pound he need fear no trouble from steel fittings and steel valves. Steel at four cents or five cents per pound has not always proven a success and a reliable material of reasonable price is desirable for a great deal of steam pipe work. A high grade cast iron is capable of withstanding the demands made upon it by superheated steam, and fittings made of this material have been in successful use for the past five years. The paper describes the experience with these fittings.

B. R. T. Collins. Last summer I ran across three valves on pipe lines from boilers on the main steam header so located as to be subjected to excessive expansion strains as described by Professor Hollis. They were 10 in. extra heavy valves, with ribs running between the end flanges and also between the bonnet flange and end flanges. These latter ribs were cracked from 1 in. to $1\frac{1}{2}$ in. deep on all three valves. In addition one valve had a crack 1 in. deep in one of the longitudinal ribs, and in one place on the body showed small criss-cross cracks when examined with the miscroscope.

2 The face-to-face length of this valve was originally 18 in. but after two and one-half years' exposure to a superheat of 150 deg. this had increased to $18\frac{7}{32}$ in. This valve was removed, broken up, and pieces sent to Professor Miller for testing, which showed a tensile strength of 11,300 lb. per sq. in. The iron was very coarse, with crystals something like $\frac{1}{8}$ in. across. This valve evidently was of very poor material to start with, or else it was seriously affected by expansion strains due to its location or to the superheat. Probably all three of these conditions had their share in producing the result obtained.

George A. Orrok. When we first considered the use of superheated steam in our power stations a few years ago there had been developed a type of steam piping which most engineers considered excellent. The piping itself was of steel with VanStone flanges, the flanges being of sufficient thicknesses to prevent buckling. The fittings were all cast iron of a carefully worked out pattern, much stronger than the ordinary high-pressure fittings. The valves were of similar design and the whole piping system was bolted together with steel bolts of larger size and greater number than the ordinary extra heavy standard required.

2 This piping system gave absolutely no trouble with saturated steam. The up-keep of such a system under power station conditions with 200 lb. steam pressure over a period of a number of years was almost nothing; in fact less than $100 was spent on one pipe line in about three years time.

3 Superheated steam, however, introduced another factor, and a very important one. From certain tests made by the General Electric Company it was considered that this superheat might vary over a range of more than 200 deg. and the temperature strains brought upon the piping and the valves would be severe. It was finally determined to make the entire pipe line of steel. The prices on

steel valves and fittings were only a little higher than if of a good quality of cast iron of the thickness required for the high pressure and excessive temperature strains. We adhered to the steel piping with the VanStone joint, but made the VanStone flange of cast steel from the cast iron pattern. The steel fittings were not as heavy as cast iron ones of the same size but differed considerably in the detail. of design. The steel valves followed the design of the fittings and were of various makes, both single and double wedge.

4 Our experience with the steel valves has been good and we feel that they are giving better satisfaction than was to be expected under the circumstances. Troubles developed from blowholes, however, which led to an investigation of the subject about a year and a half ago. We traced most of the blowhole difficulties to improper moulding, improper gating and to over-oxidized metal in the case of Bessemer steel and cold metal in the case of open-hearth steel. The valves and fittings are about equally divided between Bessemer and open-hearth steel, all of the former being made, however, on the baby converter by the Tropenas and Zenzes process. The manufacturers understand better today how to handle the work, and the castings which we are receiving are much better than they were two or three years ago. I believe our troubles with the steam lines resulting from superheat are now practically over, and on one steam main in particular we have done nothing in a year and a half. Whether or not the valves can be shut off absolutely tight I do not know as we have had no reason for doing this during the time.

5 After our first installation of steel fittings and valves I had occasion to look up a number of power stations in which cast iron fittings and valves had been installed for use with superheated steam. In one of these stations I saw fittings which had been under the action of superheat for approximately nine months and had been removed because of the many leaks which had developed. The castings were supposed to have been made from the best air furnace iron, but were swollen and bulged practically all over, the outside being covered with fine hair cracks. None of the castings had gone to pieces but practically all had developed leaks, and were being replaced with steel.

6 At another station a cast iron valve had gone to pieces causing quite a little damage and many other valves and fittings had been seriously affected. There were a number of vertical engines in this station in which superheated steam had been used. All the high-pressure cylinders had cracked in two or three places and they were

replacing the cylinders and had so arranged their pipe line that no more superheated steam could get to them. The fittings which I examined, taken from the superheat line had all undergone a growth in size and the outside was covered with fine hair cracks and seemed very much swollen. Analyses of the metal showed a silicon content of from 1.88 per cent to 2.33 per cent, phosphorus about 0.7 of 1 per cent, low manganese and almost no combined carbon. The tensile strength of the material after its exposure to superheat was in the case of the iron with the silicon content of 1.88 per cent about 4500 lb. per sq. in.; in the case of the silicon content of 2.33 per cent it averaged about 8500 lb. per sq. in. We have no means of knowing what this was when it was first made. Microphotographs of the etched surfaces of this metal show the essentially open character of the iron. In this particular station the superheaters have been removed and their troubles have ceased.

7 In view of the many and excessive strains likely to come on a pipe main with 200 lb. pressure and more or less superheat I have not felt that we are justified in installing cast iron valves and fittings. Even with saturated steam at the above pressure and with the length and size of mains which we are using today in our modern stations it seems to me that the extra expense for steel is justifiable and might probably be saved many times over in the cost of up-keep during the life of the station.

8 A few years ago it was the general impression that superheated steam could not exist in the presence of water. This statement has been made many times and no longer ago than at the Annual Meeting. That this idea is fallacious is, I think, the generally accepted belief today, and we have good evidence that it is possible in a steam pipe carrying steam at 200 lb. pressure and 200 deg. superheat to have a stream of water flowing along the bottom of the pipe. In this case the bottom of the pipe would be at a temperature of possibly 380 deg. fahr., while certain other portions of the pipe in contact with the superheated steam might have a temperature between that of saturated steam and the maximum temperature of superheat.

9 Regarding the difference between European cast iron and American cast iron, it has been my impression that the pig-iron manufacturer here is always trying to make a grade of iron which will command a high price in the market. This iron must be an open iron with reasonably high silicon and almost no combined carbon, the carbon content being in the graphitic state. This iron

will sell readily. If the quality of the iron fell off, and because of a lower silicon content more of the graphite was converted into combined carbon, the iron would become harder—more difficult to machine—and would not command as ready a sale. In Europe, it is my impression that they make much harder iron and are willing to spend the money to machine it. In America we demand an open iron that can be machined easily.

10 If Mr. Mann continues his researches and considers his test specimens in the light of the volumetric composition of the iron; that is, the volume which the compounds of iron and silicon and of iron and carbon occupy in the cast iron, in comparison with the volume occupied by the iron itself, he may find some interesting results.

11 Referring to air-furnace iron, or gun iron as it has been called, I think the great difficulty is the fact that it is almost impossible to control the regular composition of the product. The reverberatory furnace, while a comparatively simple piece of apparatus, is remarkably delicate, and uniform results are obtained only when the very best of care is taken. It is a comparatively easy thing to refine high silicon iron to some kind of refined iron, but it is a much harder thing to get a uniform result from each heat.

W. K. Mitchell. The following notes are taken from several years' personal experience with superheated steam and its effect on cast-iron valves, fittings, etc.

2 Our first intimation that cast-iron fittings and valves gave trouble under superheated steam conditions occurred about three years ago and came in the nature of a surprise, as we had been using superheated steam for some years previous.

3 The first case was in a railway power plant for a high-speed electric line. The plant had been running for several months under a fairly constant load, but owing to a falling off in traffic it was decided to cut down the service to one-half or less, which made the load quite variable. Three months after this had been done the trouble with the fittings and valves began to develop. It was first found that the valves could not be closed tight, and gaskets were giving trouble. Then fittings began to show signs of weakness, cracks appearing on the outer surface.

4 Fortunately these cracks never extended through. In an 8 in. by 6 in. double tee, the metal of which was about $\frac{7}{8}$ in. thick, the cracks did not extend more than half way through, which would

indicate that there is no advantage in very thick castings under such conditions. The most serious of these cracks occurred at the junction of the flange and fittings, and kept growing to so alarming an extent that several fittings were replaced. It was then noticed that the old fittings had lengthened considerably. The original length of some 8 in. by 6 in. double-tee fittings was 35 in., and when taken out and cooled they measured $35\frac{5}{8}$ in. to $35\frac{3}{4}$ in. They had been in service about nine months. Open hearth cast-steel fittings and valves were substituted for those of cast-iron materials and have been working satisfactorily ever since.

5 About the only information we could get bearing on the cause of this growth was from a paper by A. E. Outerbridge, read before the mining and metallurgical section of the Franklin Institute in January 1904. Mr. Outerbridge stated that by repeated heating and cooling of bars he had caused the metal to grow to an almost incredible extent. He exhibited a test bar, the original dimensions of which were 1 in. square cross-section and 14 $\frac{13}{16}$ in. long, which had been heated some 27 times to a temperature of about 1450 deg. fahr., and cooled again by various methods, some slow and some fast, until at the end of the treatment it had grown to a length of $16\frac{1}{2}$ in. and a cross-section of $1\frac{1}{8}$ in. square. The similarity between the action of the fittings above referred to and the test bars which Mr. Outerbridge exhibited caused us to investigate further along similar lines.

6 The railway plant was designed for a steam pressure of 175 lb. per sq. in. and superheat was intended to be 150 deg. fahr. That this temperature had been greatly exceeded, however, was made evident by the discovery of a board that had been charred by contact with the steam trap which rested on it. This trap was connected to a drip pipe running from one of the elbows next to the strainer on a steam turbine and was about 10 ft. below the elbow. Investigation showed that the trap had been so hot that its legs had burned holes through the board until the trap was not resting on the board at all but was suspended by the pipe.

7 The president of the company that installed the superheaters said that while they were built to give an average of 150 deg. superheat, "the real question was not one of the amount of superheat but of velocity." This seems reasonable when one considers that if the load should fall very low, the velocity of the steam through the superheaters would be considerably reduced and its temperature correspondingly raised. Again, a sudden increase in the demand for steam

would result in a rapid flow through the superheater, and steam at much lower temperature, and these recurring changes of temperature must necessarily cause rapid changes in the lines due to expansion and contraction. We therefore concluded that in this particular plant, at least, the damage to the fittings and valves was not caused by the high temperature itself, but by the constantly changing temperatures due to the change of load.

8 Our contention that the damage was due to variable temperatures seemed to be borne out by the fact that in a cotton mill plant installed three years previously, where the steam requirements for pressure and superheat were higher than those mentioned (the pressure being 200 lb. per sq. in., and superheat 200 to 250 deg. fahr.), the fittings and valves were of regular cast iron, and there had been no trouble to speak of. The load was practically constant, however, varying not more than 15 per cent at any time.

9 On account of the discussion in several publications in the spring of 1908 regarding the disastrous effects of superheat on cast iron, the owners of the mill grew anxious about their piping and asked the writer to look over the system. He found everything normal; the fittings were tight, valves could be operated freely, and in a general way the plant was in good condition. The first installation had been made in 1903, and a second one in 1906, using the same class of fittings and valves.

10 The writer suggested that measurements be taken of all the fittings and valves in the plant and records kept of changes. The original dimensions were determined as closely as possible from the patterns and records of construction, and beginning with July 1908, records were kept of the dimensions of the fittings for a period of nine months. Although the changes in the dimensions were slight, the increase in length of certain of the fittings was such that it was thought unsafe to continue them in use and steel fittings were substituted throughout. Most of the valves, however, are still in service and there have been no failures in either fittings or valves. The following will give an idea of the changes which occurred from the dates of installation to the last date given:—

A 12 in. by 10 in. by 8 in. by 6 in. cross installed in 1903 measured 24. in. in length, and in March 1909 measured $24\frac{21}{64}$ in.

A 10 in. by 8 in. by 8 in. tee increased in the same time from 24 in. to $24\frac{5}{16}$ in. and

A 12 in. by 12 in. by 8 in. by 6 in. cross, 20 in. long when installed in 1903, was $20\frac{9}{64}$ in. long in March 1909.

A 12 in. by 10 in. by 8 in. by 6 in. special cross, 50 in. long when installed in 1906, had grown to $50\frac{1}{2}$ in. in 1909, also

A 10 in. by 8 in. by 8 in. tee, $24\frac{1}{2}$ in. long, measured $24\frac{47}{64}$ in. in March 1909.

11 These facts seem to show that even at high steam temperatures, if cast iron can be kept at a uniform temperature and not cooled off too frequently or too rapidly, it will meet the requirements of superheated steam for a long period; but if the temperature is subject to frequent changes such as occurred in the railway plant, the cast iron will become disintegrated and ultimately fail within a short period of time.

12 In another street railway power house which had been in operation for a number of years with saturated steam at 200 lb. pressure, cast-iron fittings, valves and pipe were used successfully. During 1906 fourteen new boilers were added, making a total of thirty-two. The new boilers were equipped with superheaters intended to superheat to about 50 deg. fahr. New piping was installed similar to the old, with cast-iron fittings and valves, and steel pipe with steel flanges. The fittings were unusually heavy and strong. In the original installation of this piping a white metal gasket had been used, about $\frac{1}{16}$ in. thick, which was very satisfactory for saturated steam. These gaskets had a melting point of about 650 deg. fahr., and no sooner had steam been turned in from the new boilers than the gaskets began to melt, and in the course of a month or six weeks it became necessary to replace every one with material that would stand the temperature of the superheated steam. As the majority of the boilers had no superheaters it was hard to understand how sufficient superheat could be generated by the new boilers to do any harm.

13 Two years later a 16-in. tee in one of the connecting pipes between the main headers was found to be leaking. The leak becoming worse, the covering was taken off and the tee was found to be covered with small cracks or fissures similar to the cracks that had occurred in the fittings taken out of the power house first mentioned, except that a few of the fissures had worked through to the inside. The tee was replaced with a new one of the same material and dimensions. When the defective fitting was examined it proved to be something of a curiosity. Its original dimensions were 31 in. face to face by $15\frac{1}{2}$ in. centre to face. It had grown on one side to $31\frac{7}{8}$ in. and on

the other side to 32½ in. The flanges, which were originally 25 in. in diameter had grown to 25¾ in., and as they had been bolted to steel flanges that had not changed under the superheat conditions, they had become dished to a depth of about ⅛ in. The original thickness of the body of this fitting, as near as could be determined from the pattern, was about 1¾ in., but where the surface cracks were most numerous careful measuring gave a thickness of almost 2¼ in. Of course, there is always the possibility of the core moving when a casting is being made, but the thickness of this fitting was quite uniform throughout.

14 In this plant, the trouble did not stop with the fitting. Several valves began to show cracks and were replaced. Then the high-pressure cylinders of the engines became affected, and several had to be renewed. The engineers finally decided to take out the superheaters, and the plant is now running without superheat. This is a typical street railway plant, subject to changes of temperature similar to the one first mentioned. Fig. 1 shows the original dimensions of the 16-in. tee and its dimensions after coming out of the line.

15 It is of interest to note that at the same time these superheat boilers were installed in the railway plant, similar boilers with engines, piping, valves and fittings of the same type and material, were installed in a lighting plant in the same city, where practically no trouble of any kind had developed. This seems to indicate that a much more constant load is maintained.

16 It is my opinion that in plants where the load is constant and the temperature of the steam therefore constant, properly designed piping with cast-iron fittings of good material will do the work satisfactorily and be safe for a long time. I believe there is no advantage in using cast-iron alloys known as semi-steel, ferro-steel or gun iron. In the railway plant first referred to, the fittings were of cast iron from one foundry; gate valves of semi-steel from another; and stop, check and emergency stop valves also of semi-steel from a third. The results in each case were practically the same. A specimen from an 8 in. by 6 in. double tee which had been in service about nine months gave a tensile strength of 13,750 lb. per sq. in. The chemical analysis of this piece gave the following: Carbon, 2.502; Phosphorus, 0.461; Sulphur, 0.083; Silicon, 2.435.

17 Two test pieces from the 16-in. tee showed tensile strengths of 4970 lb. and 4340 lb. Chemical analysis: Silicon, 2.33; Sulphur, 0.07; Phosphorus, 0.68; Manganese, 0.39; Total Carbon, 3.18.

18 In Fig. 1 is shown the first fitting listed in Par. 10, taken from

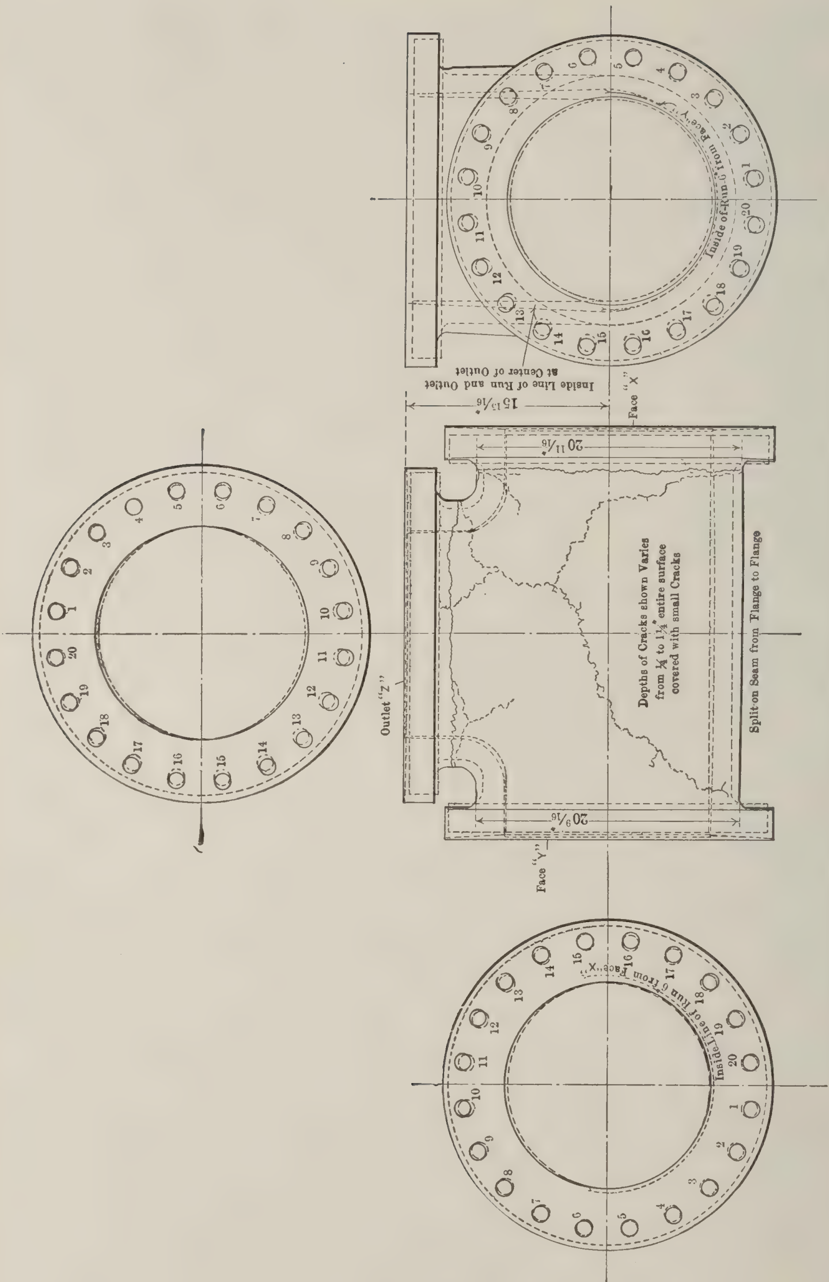


FIG. 1 CHANGES IN DIMENSIONS OF 16-IN. TEE REMOVED FROM STREET RAILWAY POWER STATION

the cotton mill plant where it had been in use several years under superheated steam, the temperature of which, however, was nearly constant. This fitting was tested under hydraulic pressure. At first the pressure was put up to 1100 lb. per sq. in., when the gaskets leaked and the pressure had to be reduced to zero in order to tighten up the gaskets. The fitting was then tested again and broke at a pressure of 1250 lb. There were no serious defects in the fitting. One small surface crack was of so little moment as not to

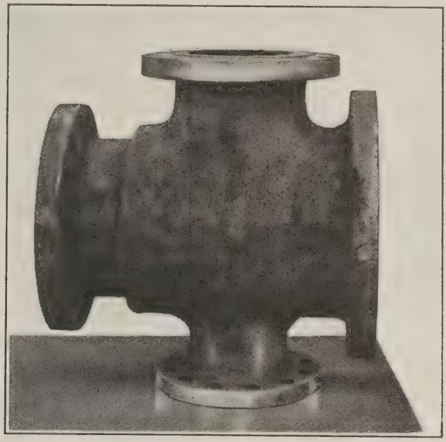


Fig. 2 12 in. by 10 in. by 8 in. by 6 in. Extra-Heavy Cast Iron Fitting in Use with Superheated Steam, 1903–1908

require special attention. A testpiece from the fitting showed a tensile strength of 15,900 lb. per sq. in. The chemical analysis was as follows:

Total carbon	3.05
Phosphorus	0.769
Sulphur	0.06
Silicon	2.07

19 It will be noted that the silicon in this specimen is lower than in the two castings just mentioned and I think this had a good deal to do with the case, as well as the fact that the load and temperature were constant.

20 While open-hearth steel castings seem to be successfully used under superheated steam conditions, I do not believe they will last indefinitely because of their extreme thickness. There must be

changes taking place similar to those in cast-iron, due to temperature changes, but the ductility of open-hearth steel will undoubtedly delay the process of disintegration for a longer period. The material which we recommend and use today for high-pressure superheated steam, is wrought steel throughout, with welded nozzles instead of fittings, and steel flanges, using bends in all cases in preference to short elbows.

John Primrose. During the past eight years the writer has been in close touch with many plants, containing upwards of fifteen hundred installations using superheated steam, and in a position where troubles would be promptly reported to him. Almost without exception these plants use cast-iron fittings in their pipe connections. The fact that no one of these plants has reported troubles with its fittings is in striking contrast to the comparatively few instances where superheated steam has been charged with being the cause of trouble with cast-iron fittings. In order that there should be no doubt about the absence of trouble due to superheat, letters were written to ten concerns known to have been passing superheated steam through cast-iron fittings for the past eight years, at from 100 to 150 deg. superheat, asking the following questions:

Question One. Are not the tees, elbows and valves of cast-iron in the branch and main steam lines leading from the boilers? Seven answered yes, two replied that some fittings and valves were of cast-iron and some of cast-steel, and one replied that while the fittings were originally of cast-iron some tees had been changed to cast-steel, but stating positively that the change was not made because of any ill effects of superheated steam.

Question Two. Are fittings of extra heavy or standard weight? Nine replied that they used extra heavy fittings, and one standard weight.

Question Three. What steam pressure do you ordinarily carry? One used steam pressure of 100 lb., six used 150 lb., one 165 lb., one 185 lb. and one 200 lb.

Question Four. Have you ever noticed any injurious effect of the superheated steam on valves or fittings? Eight answered no, one that no trouble was experienced in fittings, but that valves with cast-iron bodies and brass seats were difficult to keep tight, and one reported no trouble further than the baking of a hard deposit on inside.

Question Five. Have you ever found it necessary to replace

any of these valves or fittings with cast-steel? Eight answered that no fittings or valves had been replaced on account of superheated steam. One answered that they had replaced no fittings, but some globe valves, and one answered that they were replacing some fittings with cast-steel, but upon further inquiry it was found that this was not because of the ill effects of superheat, but because the steam mains were being changed to contain VanStone joints and they wished to change the fittings to standard length and deemed it advisable to use cast-steel.

Question Six. Of what material are the gaskets in the steam line? Seven use corrugated copper or bronze, two sheet packing, and one asbestos.

2 The chief engineer in charge of a plant in the middle west, of some 20,000 h.p., writes that nothing has developed in any of the cast-iron fittings to show that they are in any way affected by the use of superheated steam. This plant has been in operation about five years.

3 Such evidence as the foregoing proves pretty conclusively that superheated steam does not have an injurious effect on cast-iron. There seems to be no very good reason why it should. There is nothing extraordinary in the fact that several cast-iron fittings have failed when passing superheated steam. The failures were probably due to inferior metal, or to strains developed by expansion or contraction of the pipe lines, as suggested by Professor Hollis and Mr. Mann. These are much more plausible theories than that superheated steam at a temperature of 500 deg. to 600 deg. fahr. has any effect on the metal. In investigations by Mr. Outerbridge and Professors Rugan and Carpenter on the growth of cast-iron when repeatedly heated, their experiments were started at 900 deg. C. or 1652 deg. fahr. Such instances as the growth of grate bars, etc., are all at temperatures far exceeding anything used in superheated steam work for power plants.

4 Samples of cast-iron taken from fittings passing superheated steam for years have been polished and micro-photographed before and after etching, and compared with samples treated in the same way, taken from fittings passing saturated steam. The report states that there is no evidence of a change in the carbon conditions, or of exposure to superheated steam, and in support of this a well known foundryman gives his opinion that a temperature below 900 deg. fahr. would not produce any effect in cast-iron.

5 The tests of the famous Crane valve so often quoted are no proof of superheated steam being responsible for the failure. Test bars from the broken valve were compared with test bars taken from the same heat that the valve was made from, and the valve was said to have weakened. This is no real test, because castings from different parts of the same heat, or, in fact, different parts of the same casting are known to vary in strength, and it is quite likely that fittings passing saturated steam, if compared on the same basis, would be found to have suffered greatly from the effect of saturated steam! It is unquestionably true that this valve must have been subjected to other influences besides superheated steam. It is rather remarkable that the body of the valve is said to have been weakened more than the flanges—the reason given being that the metal of the body was nearer the superheated steam. Is it not more reasonable to suppose that the metal of the body weakened more than that of the flanges because it was subjected to greater fatigue on account of expansion and contraction of the pipe?

6 The writer's experience with a great number of steel fittings used for pressure parts of superheaters exposed to hot gases, has led him to conclude that the metal in steel-castings is anything but satisfactory for fittings. It unquestionably has greater tensile strength than cast-iron, which appears to be its only advantage. On the other hand, it is difficult to get steel castings sufficiently homogeneous to hold the pressure. A large percentage of castings are "doctored" before leaving the foundry, but new openings frequently develop on test after machining, and even after the castings are in place, causing the charge to be made that the castings have not been tested before sending out. This fact is further evidence of strains developing in service, other than those produced by internal pressure, which open up cavities or spongy places not discovered by shop test. Steel castings vary greatly in toughness, as shown by the great variation of elongation on test; others are so hard that machining is very difficult. While they can be bought on very careful specifications to guard against these faults, there is always the chance of porosity. The high tensile strength of the steel is not a necessity, and cast-iron made to careful specifications is amply strong. It machines well, is not porous, and can be relied on to hold the pressure.

7 Care should be exercised in the design of pipe lines to guard against straining the fittings from movement of the pipe due to expansion and contraction. Where long radius bends are the means of taking up this movement, pipe of the lightest possible weight consistent

with safety should be used, thereby lessening the force required to spring the pipe. With properly connected flanges, full weight pipe is amply strong for all ordinary working pressures, and if drawn tubing is used, even lighter metal may be adopted. In this connection the design and arrangement of the superheater is of great importance, and should be such that sudden and frequent changes in the temperature of the steam do not occur; otherwise the changes in the length of the pipe will be more frequent, resulting in a more rapid fatigue of the metal of the fittings.

8 A better way of taking care of expansion than with long radius bends, is to use ball and socket expansion joints, which have the additional advantage of reducing the amount of piping.

9 The writer agrees with Professor Hollis in charging strains due to expansion and contraction with the failure of certain fittings, and with Mr. Mann when he charges inferior fittings with the cause of failure in other cases and recommends the use of a good cast-iron containing a percentage of steel scrap for fittings passing superheated steam. This is entirely in accord with the writer's experience.

H. S. Brown believed the troubles with cast iron would be eliminated if the temperature could be kept constant, and further said:

2 I think the discusssion boils down to this, that under certain conditions steel castings will give a more satisfactory performance than cast iron. The company with which I am connected has found it necessary to replace a large number of cast iron fittings with steel castings, where superheated steam was used; and the performance of these steel fittings, under the same conditions under which the cast iron fittings were working, has been satisfactory.

3 In a large number of other plants where cast iron fittings are used with saturated steam, the design of the piping is such that the stresses set up on account of expansion and contraction are very much worse than in this system; and we do not get into troubles as we did in using superheated steam. It may be that steel fittings will form a more practical and less expensive way of taking care of the conditions set up by the use of superheated steam than elaborate precautions in the way of expansion joints and the like.

E. H. Foster. Cast-iron is much too useful a metal to receive general condemnation for steam pipe fittings, whether for superheated or saturated steam, without very good reasons and the writer is firm in this opinion that such reasons have not yet been advanced.

2 Having devoted the greater part of his time for the past ten years to the study of superheated steam and the manufacture of superheaters, the writer has eagerly followed up every report of the failure of a steam pipe fitting, where superheated steam was used, and it can fairly be said that no instance has yet occurred where the weakness has not been readily explained by the poor quality of the iron, or by lack of provision for expansion and contraction without straining the metal. The many instances where cast-iron fittings are habitually subjected to steam of varying degrees of superheat up to final temperatures close to 1,000 deg. fahr. leave no doubt that good cast-iron is equal to, if not better than, any other metal for making steam fittings for superheated steam as well as for saturated steam, especially in smaller sizes.

3 In the writer's experience it is as important to have regard to the mixture of the iron to be used in cast-iron fittings as it is to secure the proper mixture for concrete work.

4 The suggestion that better provision should be made for free expansion and contraction of steam pipes, is, in my opinion, very much to the point. More care applied to this feature of the design of power plants would remove entirely from the shoulders of cast-iron the odium of being unsuitable for carrying superheated steam.

L. B. Nutting stated that superheaters installed by his company nine years ago, and since then in constant use, have caused no trouble and have not changed their dimensions. These superheaters were made entirely of cast iron, the tubing having a smooth bore and corrugated exterior.

2 He also reported a great many superheaters installed and in operation delivering steam at a temperature of 1000 deg. fahr. On these the users have employed, without any distortion or without any evidence of weakness developing, standard makes of cast iron valves (globe valves and angle valves) under 1000 deg. final temperature. But the temperature is maintained at 1000 deg., without a variation of 25 deg. These illustrations seem to trace the cause of the trouble with cast iron fittings directly to widely fluctuating temperatures.

3 In regard to Mr. Mitchell's suggestion that provision should be made to obviate troubles from varying temperatures on cast iron Mr. Nutting asked, why not make provision to keep the temperature constant. The art of superheater construction has advanced to such a point that a uniform temperature should safely be counted

on. The plant Mr. Primrose referred to, a 20,000-h. p. boiler plant has a record of variation not exceeding 10 deg. either way from the desired amount at any time during the year, although the loads had a fluctuation of from 5000 to 35,000 kw.

ANDREW LUMSDEN.[1] In one case we have eight boilers of the Babcock & Wilcox type, equipped with heaters part of which were made and installed by the boiler company. These boilers were all connected to one 12 in. main through long radius bends, valves, tees, etc. On the superheaters installed by the boiler company there were usually 150 deg. of superheat and on the others about 90 deg.

2 When this plant had been in service about two years some of the fittings in the main were found to leak just back of the fillets and small cracks were discovered extending around one side of the tees. Some long bolts were made to go the whole length of the tees and through the end flanges, using them to make the joints. Steel tees were also ordered of the same dimensions as the cast-iron ones to replace all the fittings in the main. When the old fittings were removed, however, it was found they were from $\frac{3}{8}$in. to $\frac{1}{2}$ in. longer than when first installed. This is a turbine station and they have had quite a little trouble with the admission valves on some of the turbines, those directly opposite the boiler carrying the 150 deg. superheat giving by far the most trouble.

3 At another plant there are boilers of the Babcock & Wilcox type and Curtis steam turbines, installed seven years ago, with a separately fired superheater on which exhaustive tests were made. The temperature of the steam leaving the superheater reached as high as 750 deg. The superheater was run for about six months and at the end of that time all the copper gaskets in the main were destroyed and the joints had to be remade. The superheater was shut down and has not been operated since.

4 The writer visited this plant a few days ago and found that no large joints had been made since the superheater was shut down. There are about ninety joints ranging from 8 to 12 in. diameter and none have leaked, but three 12-in. valves were leaking badly through the body on the under side and about the point where the seat rings are screwed in. These valves are laid on their sides and are on the boiler side of the superheater and have never had superheat in them nor in

[1] President, Lumsden and Van Stone Co., 69-71 High Street, Boston, Mass.

the writer's judgment can their troubles be due to expansion as particular care was taken to allow free movement in all the piping of the plant.

5 At another plant where they have Babcock & Wilcox boilers, Curtis turbines, etc., they have carried 150 lb. steam pressure and 150 deg. of superheat for about six years, with cast iron fittings, etc., and have had no trouble with the fittings. The valves have given them some trouble with loose seat rings and by being badly cut, and some of them have been replaced.

John C. Parker. Professor Hollis draws attention to the necessity for greater allowance for expansion in piping for superheated steam. This is important and where sufficient flexibility cannot be put into the design expansion joints should be installed. My experience accords with his statement that cast-iron fittings have been largely and successfully used for superheated steam.

2 Six or seven years ago I was called on to furnish superheaters with some of our boilers but could find none in the market to meet my ideas of what a superheater should be. A design was worked out and forty or fifty thousand horsepower of boilers have been built with these superheaters. Two plants are above ten thousand horsepower. In six years experience we have had no trouble either with cast-iron or steel fittings, valves or cylinders. I ascribe the result to the steadiness of the superheat and to the fact that condensed steam in the superheater is not intermittently carried into the steam line.

3 In one of our first installations the men started to flood the superheaters without my knowledge whenever the boilers were banked and the result taught me the effect of suddenly injecting water at 360 deg. fahr. into piping and headers which had been raised to 500 deg. fahr. Leakage started at the joints but stopped as soon as the practice was stopped.

4 I believe there is no connection between the troubles which I have been cognizant of with some designs of superheaters and the expansion of the steam mains. I believe the troubles have been due solely to fluctuations in temperature and temperature shocks caused by frequent injection of condensed steam from incorrectly designed superheaters. I recently went into a plant where a superheater had been in use for about four years. It was an independently fired superheater and the engineer had had so much trouble with piston rings in an engine with poppet valves designed especially for superheated steam, that he had cut the superheat from 600 deg. fahr. to

500 deg. fahr. and then to 400 deg. fahr. He had his ideas centered on the point at which the trouble occurred (the cylinder and rings) and would not believe that it was due to fluctuations of temperature. I finally pinned him down to the statement that "the temperature could be controlled perfectly but you had to watch it like a cat."

5 There is a superheater in the market that uses cast-iron to protect wrought iron tubes. I have seen some of these removed from boilers on account of overheating and, while the cast-iron had been red hot it had cracked less than some steam pipe fittings which had been subjected to water jets and fluctuations under 600 deg. fahr.

6 I do not think conclusions can be drawn from Professor Miller's experiments. It would require at least half a dozen tests of the same sample of cast-iron at progressively increasing temperatures and periods to obtain results of value. Ten of the tests show increased strength of cast-iron while all the steel has lost strength. One steel test (100,000 lb.) is unreasonable.

7 It is unfortunate that Mr. Mann has not given us plans of boilers and superheaters and steam piping, and such data regarding the conditions under which the fitting troubles occur, as would permit more intelligent study. I note that steel fittings have failed in some cases. This would indicate water jet action, since we have never had to renew a single steel header in our superheaters.

8 We have sixteen 800 h.p. boilers with superheaters directly over the fire running at 175 lb. pressure and up to 170 deg. superheat with no such trouble as Mr. Mann mentions in Par. 12.

Albert A. Cary said that after investigating a number of plants having trouble with the use of superheated steam, he had been led to the conclusion that many if not most of their troubles have been due to bad design in the piping arrangements.

2 Far greater care and better judgment is called for in designing pipe systems for superheated steam than for similar systems using saturated steam, as the strain due to expansion and contraction is greatly increased. The piping on each side of every offset should be carefully considered to see that excessive stress is not thrown upon the flanges and threads by the lever which is developed there. Several special forms of flanges which avoid the screw connection are now used to excellent advantage, with high superheat.

3 Continued flexing on one side of the flanges of fittings, due to the cooling and high degree of heating of the pipe system as steam is turned on and shut off will cause ruptures not unlike those shown in

the illustrations accompanying these papers and will be apt to change the internal structure of the metal itself.

4 As higher velocities are permissible, in pipes carrying superheated steam, than in those for saturated steam, somewhat smaller piping can be used to excellent advantage, not only decreasing the cost, but also increasing the flexibility of the pipe line.

5 Considerable care must be exercised in flexible pipe bends, which should be bent to large radii, and placed in the most effective positions with a proper anchoring of the pipe on either side of the bends. Careful supervision must be exercised in seeing that the pipe fitters make up their pipe lines in such a way as to avoid any severe springing of the pipe in order to make the flanges of the joints register properly, one against the other. Undoubtedly, neglect in these particulars of pipe design and pipe fittings is responsible for a very large percentage of the failures found in the fittings and joints in superheated steam plants.

6 I have been led to consider the handling of superheated steam in power plants under two headings, first superheated steam having a total temperature below 500 deg. fahr.; second, superheated steam above this temperature. The first-named quality of steam, with a total temperature below 500 deg., will be found better applicable to old plants, in which it may be newly introduced, and in many new plants. With such steam we may use the ordinary high-grade fittings used in saturated steam work, with some modifications.

7 The high-temperature superheated steam in power plants is more especially adapted to installations where high-pressure steam is required for steam turbines.

8 Mr. Cary referred at some length to the possibility of cast iron being affected by heat under stress, calling attention to the discoveries under the microscope of the change in physical condition of alloys produced by heat treatment, and advocated the carrying out of experiments along these lines.

W. E. Snyder. I may be able to touch upon certain particular phases of the papers and the discussion, in such a way as to contribute some of the results of actual experience covering a wide variety of conditions and several years' practice. The consideration of this subject in the discusssions seems to have broadened to include the effect of unequal heating of metal and also the designs of systems of steam piping. These matters are both directly related to the use of cast iron fittings for superheated steam, as failures may in

some cases be due to the improper design of the steam piping; also under some conditions to unequal heating of such irregular castings.

2 A common connection between engine and main steam pipe is by a branch running horizontally and at right angles to the main pipe, out directly over the high-pressure cylinder and turning down by a bend to connect with the throttle valve on top of the cylinder. When this branch is long the expansion in the steam line does not exert any harmful effect in the throttle valve, but the expansion in the branch is taken up by the change of curvature of the bend over the throttle valve, and this puts a strain directly on this valve. In two or three instances where this kind of connection was in use, the throttle valves were cracked immediately under the flange, and serious accidents narrowly averted.

3 Where the branch to the engine is short the expansion in the branch itself does not require any consideration, but the longitudinal movement of the steam main due to its expansion and contractions, transmits strains through the branch pipe directly to the throttle valve and flange on the steam chest. In one instance this resulted in a very serious accident, as the cast iron flat top of the steam chest was broken in by the expansion of the main steam pipe several feet away.

4 In another installation a 24 in. cast iron Y with an 18 in. branch split in the fork of the Y, while under 150 lb. steam pressure. Fortunately it was possible to take the line out of service before the fitting exploded, but it was a very narrow escape. All the accidents mentioned above were the direct results of the installation of systems of steam piping without proper consideration of the effects produced by expansion and contraction. All were in systems using saturated steam, and they emphasize the necessity of using great care in arranging the piping that the expansion and contraction may take place without throwing the severe strains on the cast metal members, which are always liable to failure under such conditions. Consideration of this feature of design is of still greater importance in piping systems using superheated steam, on account of the higher temperature used and the consequently greater expansion. The avoidance of expansion strain on castings in a system of steam piping is of fully as great importance as is the selection of the material from which these castings are made.

5 The effect of unequal heating of metal has been investigated by engineers in the French Navy (See Marine Boilers by Bertin &

Robertson, p. 201). The theory advanced there, which seems reasonable and is also confirmed by experience, is this: When one side of a piece of metal or a boiler tube is heated to a higher temperature than the other, the hot side tends to expand, and the expansion is resisted by the metal on the cold side. This condition puts the metal on the hot side in compression, and the metal on the cold side in tension, and if the temperature difference is great enough the metal will be strained beyond the elastic limit. When the hot side is allowed to cool it is shorter than the cold side because of the strain beyond the elastic limit which has been undergone by both sides. This results in the piece taking a permanent set, or becoming "bow shaped" away from the side that has been heated.

6 The bend away from the fire of boiler tubes in some types of boilers after they have been in service for some time, seems to be a good example of the results of unequal heating. Another example is the cracking of the large cast iron mud drums used in some types of water tube boilers. Under ordinary operating conditions, in boilers having vertical baffling, the hot gas does not come in contact with the mud drum of the boilers until it has passed at least twice across the tubes, and has thus been greatly reduced in temperature. At times, however, holes are formed in the front baffle, or through the top of the bridge wall allowing the hot gases to pass directly from the furnace to the back part of the boiler setting, where they strike the cast iron mud drum, heating it to a considerably higher temperature on the front side than on the side away from the fire. These conditions have resulted, in a number of instances, in causing the mud drum to crack perpendicular to its axis, causing serious accidents.

7 This matter of the unequal heating of metal is one of the most serious with which designers of large engine cylinders have had to contend. Features of the design of pipe fittings are very similar to those mentioned in the design of gas-engine cylinders, the irregular castings having flanges and other forms of construction which make it practically impossible to avoid having the metal considerably thicker in some places than in others. This irregularity causes internal strains in the metal when heat is applied to one side. It has been the experience of European designers of gas-engine cylinders that one of the greatest difficulties they have had to overcome is this one of distributing the metal so as to avoid the small cracks resulting from irregular expansion, which destroy the cylinder. Where trouble has occurred in the use of cast iron for large fittings in superheated-steam piping, it is possible that the experience of the gas-engine engineers will suggest the remedy.

8 As bearing upon the use of cast iron for superheated steam, particularly upon the much discussed question of the possibility of having superheated steam that is in contact with water in the boiler, an experience of the speaker may be of interest.

9 A large furnace used for heating slabs for a plate mill was equipped with a small vertical Cahall boiler for the purpose of utilizing part of the waste heat. This furnace was fired with under-feed stokers, using forced blast, so that it was possible to obtain a very high temperature both in the furnace and in the boiler, also in the stack. The boiler was set in the usual way for waste heat, i.e., with the large end down. The steam pipe was connected to a flange on the top of the boiler, this connection being made inside the conical-shaped base of the stack which rested on top of the circular boiler setting. This arrangement put a cast-iron elbow on this steam pipe, and about two or three feet of pipe on each side of the elbow in the hot gas directly over the upper drum of the boiler.

10 Tests on this furnace and boiler were continued for two weeks, observations being taken every 30 minutes. Frequently the stack temperature would rise to 1000 and 1100 deg. fahr. The thermometer on a Carpenter throttling calorimeter, connected to the steam pipe just outside the stack breeching mentioned above, ranged from 380 to 600 deg. fahr., depending on the rate of working of the furnace. This variation at times occurred very rapidly, and at the time the thermometer readings were high, the steam escaping from the calorimeter was as completely invisible as though it were natural gas. For twelve hours in succession the average superheat of the steam was 140 deg. and during the entire time the tests were being made, the superheat of each 12-hr. period averaged 120 deg. or over; the steam pressure being about 95 lb.

11 This boiler has been in operation under conditions similar to the above for at least 15 years. The cast-iron elbow and flange at the top of the boiler, although subjected to such severe service as that described, has never given any trouble. A number of other Cahall boilers using blast-furnace gas, with steam pipe connections made in the same way, have been in operation for about the same length of time, and no troubles have occurred due to the fittings. The service under blast-furnace gas conditions are not so severe as the heating-furnace installation described above, on account of the stack temperature being somewhat lower, from 700 to 900 deg. The heating-furnace conditions mentioned above are unusually severe and for that reason have been described fully. It may be added that the

boilers using blast-furnace gas-produced superheated steam, notwithstanding that the water level was only a short distance below the connection to which the calorimeter was attached. (This must not be understood as being a special feature of the Cahall boiler, as in fact it is only incidental to its operation under these conditions.)

J. S. Schumaker mentioned that in the cases cited no difficulty had been experienced with the fittings of superheaters, cast iron or otherwise, but that there had been a great deal of difficulty with steam-pipe fittings. This seems to be the result of high temperature on one side of the fitting only. In the superheater itself the temperatures are balanced, to some extent at least.

Dr. D. S. Jacobus. In a large power plant that I have in mind, where the fittings are all of cast-iron, and where the superheat averages 150 deg. fahr., repeated examinations have failed to reveal any deterioration. In other cases, however, where there has been less superheat, and even where a single boiler with superheated steam has been connected into a common main with a number of other boilers furnishing saturated steam, there has been every indication that a small amount of superheat has had an injurious effect. It therefore seems that a difference in the quality of the cast-iron may affect the results, and by making a careful study of the matter and knowing the analysis of the cast-iron there is a possibility that its action under superheated steam may be predicted. In the meantime, we are furnishing cast steel fittings for all superheated steam work, as we do not know of a single case of the failure of such fittings that can be attributed to the action of the superheat.

2 The stresses due to expansion, as pointed out by Professor Hollis, may tend to produce failures. In the case of fittings broken in superheated steam lines we have found there was a stress at the point of rupture entirely apart from the stress produced by the steam pressure. In the ordinary flanges the tension of the bolts produces cross strains and the fittings give way where they would naturally fail through this strain. We have given considerable thought to the construction of flanges in which such cross strains are eliminated, but have not pushed the matter forward as we have decided to eliminate all doubt as to the safety of the fittings by employing steel castings.

3 Professor Miller's tests bear out what we have observed regarding the different results to be expected from cast-iron, as they show that although there is a general falling off in strength in one case the

cast-iron specimens did not lose in strength by being subjected to a high degree of superheat. In connection with such tests it would be interesting to investigate the action of superheat when the metals are under stress.

4 Mr. Mann's conclusion that gun iron is better than cast steel is indeed interesting, but we would not think of changing our present practice of using cast steel until gun iron is thoroughly tried out in the practical field and demonstrated all right for the work. The proper method of determining the quality of the gun metal which is used must also be developed by the necessarily slow process of observing the action of the fittings in service. It would indeed be a simple matter if bids for the fittings could be based on an analysis of the metal, and I hope Mr. Mann may be right in this belief.

Prof. H. F. Rugan. While investigating the phenomenon of the increase in cubic dimensions of cast iron as a result of repeated heatings it became evident that the test pieces deteriorated in strength. I am of the opinion that the influences at work producing such growth at high temperatures are the same that cause the failure of cast iron fittings at lower temperatures, say at from 500 deg. fahr. to 600 deg. fahr. The effect of the higher temperatures is merely to increase the extent of the changes, producing a maximum growth per heat.

2 Further experiments to determine the length of time required to produce maximum growth developed the fact that a change in the temperature was necessary to produce continued growth. No apparent difference in growth was observed between pieces heated at the same temperature for periods of 3 hours and 17 hours respectively.

3 The test pieces were heated in cast iron muffles, carefully luted with fire clay, to protect them from contact with the furnace gases, to a temperature of from 850 deg. cent. to 950 deg. cent.

4 Experiments were made with nine iron carbon alloys (A to I) containing no graphite, the carbon content changing by 0.5 per cent from 4.03 per cent to 0.15 per cent. Other constituents were low and constant. Four bars of each alloy were cast in both sand and chill moulds. These proved to be all white irons, the samples with low carbon content being full of blow holes. No growth was observed in any save the sample A which contained 4.03 per cent carbon. This sample shrank for the first 12 heats, afterwards expanding, ultimately becoming 6.88 per cent larger than its original volume.

5 Four alloys (J to M) were also tested. Of these J, K and L were grey irons while M was a white iron. It was observed that M

followed along lines closely approximating the action of A, shrinking slightly during the early heats but growing after 12 to 19 heats had been taken; ultimately becoming 6.2 per cent larger than the original. Pieces of the bars from which the A and M test pieces were made, were inserted in the muffle to be sampled for chemical analysis after successive heats. These analyses showed that the appearance of free carbon (or temper carbon) coincided with those heats which produced growth in the test pieces. Free carbon was in this way proved to be in some way an indispensable factor in the growth of cast iron when under heat treatment.

6 The grey irons J, K and L, grew from the start, and their progress indicated a close relation between their respective growth and their silicon content.

7 To check these indications a series of alloys, with all the constituents constant save silicon, having the following analyses, were used to test the part played by silicon:

Alloy	Total Carbon	Combined Carbon	Graphite	Si.	Mang.	Sulph.	Phos.
N..............	3.98	0.64	3.34	1.07	0.25	0.01	0.013
O..............	3.98	0.68	3.30	1.79	0.23	0.01	0.013
P..............	3.79	0.30	3.49	2.96	0.25	0.01	0.012
Q..............	3.76	none	3.76	4.20	0.27	0.01	0.012
R..............	3.79	none	3.79	4.83	0.30	0.01	0.012
S..............	3.38	none	3.38	6.14	0.30	0.01	0.013

8 It will be observed that the total carbon in the series is approximately constant, that alloys N and O contain about the same amount of combined carbon, that alloy P contains about half the quantity, and that the remaining alloys contain none at all. The silicon in O and R is 0.2 per cent lower, in Q, 0.2 per cent higher than was desired. The remaining constituents are satisfactorily low and constant.

9 Test pieces N to S, measuring 6 in. by about 0.88 in., were machined from the castings. They were not taken from similar portions throughout, but haphazard, some from the gate, others from a riser either near to or at some distance from the gate. When the growth of these alloys was investigated it became evident that the locations from which a test piece had been cut had a considerable influence on the rate of expansion. It was found that specimens taken from the gate end of the casting grew more rapidly than those taken from the top of the riser.

10 In test pieces from the same part of the bar, however, these inequalities disappeared and a like growth was obtained in each alloy. A slight falling off was observed in the closer-grained irons.

11 The results obtained are plotted in Fig. 1, the coördinates being percentage of growth and number of heats. In this way the rate of growth is clearly seen. In the case of samples N, O and P, curves are plotted from the data obtained. It will be observed that the growth

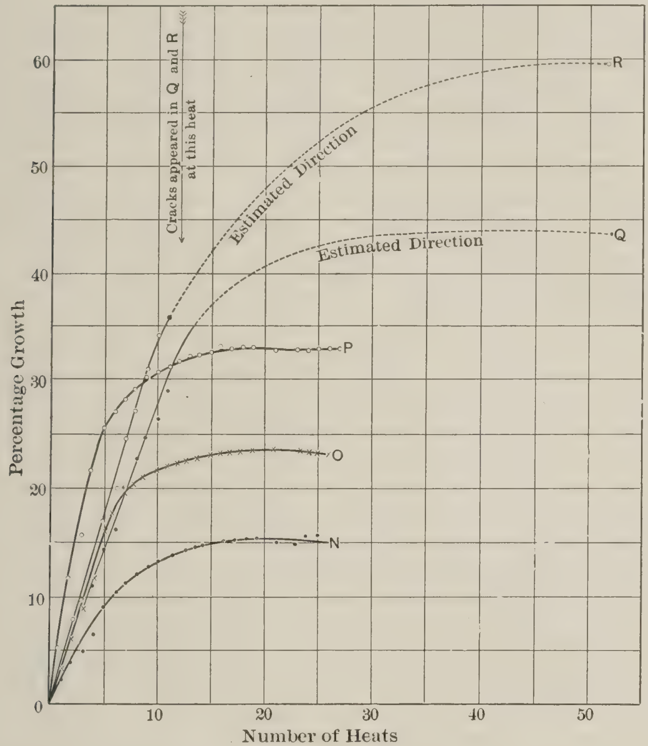


FIG. 1 CURVE SHOWING RATE OF GROWTH OF ALLOYS N TO S

is rapid at first, diminishes after about the seventh heat and stops at the sixteenth heat.

12 In the case of Q and R curves are plotted in full lines from the data obtained up to the point at which cracks appeared, viz. the twelfth heat. Beyond this the direction of the curves can only be guessed and this is indicated by dotted lines. The following table summarizes the results:

Alloy	Percentage of Silicon	Percentage Growth on Heating
N	1.07	15.40
O	1.79	23.46
P	2.96	32.85
Q	4.20	43.90
R	4.83	59.50
S	6.14	63.00

13 It is quite clear from these tests that silicon is a most important constituent of cast iron from the standpoint of growth under repeated heatings. If the ultimate growths and percentages of silicon are plotted as coördinates, the curve in Fig. 2 is obtained, which shows that, broadly speaking, the growth is proportional to the percentage of silicon.

14 To settle the question as to the influence exerted by graphite, and at the same time determine if iron-silicon alloys, containing little carbon and no graphite, would grow, three alloys (T, U and V) were experimented with, having the following analyses:

Alloy	Silicon	Carbon	Mang.	Sulp.	Phos.
T	0.65	0.17	0.17	0.045	0.017
U	1.10	0.18	0.19	0.049	0.022
V	2.71	0.19	0.20	0.051	0.033

15 Microscopic examination showed in all three cases a solid solution of iron silicide in iron. There were no traces of graphite or any other structural constituent.

16 Machined bars were heated fifteen times under the same conditions as the previous alloys. A summary of the final values is contained in the table below:

Alloy	Percentage Silicon	Percentage Change of Volume after Fifteen Heats	Percentage Change of Weights after Fifteen Heats
T	0.65	−0.025	−0.04
U	1.10	0.000	−0.03
V	2.71	+0.394	−0.02

17 It will be seen that the only alloy of the three which showed any tendency to grow was V, with 2.71 per cent of silicon. The expan-

sion, however, was very slight, and compared with that of P (2.96 per cent silicon and 3.79 per cent carbon) after the same number of heats was almost negligible, amounting to but 0.394 as compared with 31.35 per cent, the mean figure of P and PP.

18 Alloys K, N and P correspond closely to alloys T, U and V in silicon content. They also contain about 3.9 per cent of carbon,

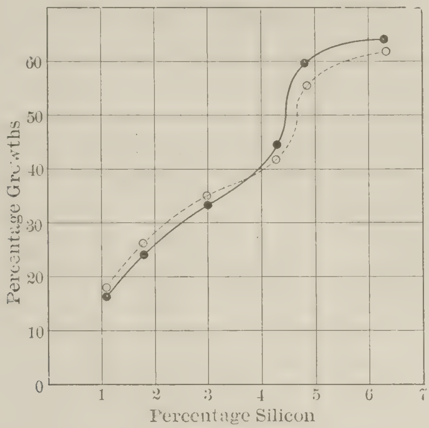


Fig. 2 Curves illustrating Relation between Percentage Growth and Percentage Silicon

mostly in the form of graphite, as compared with a mean figure of 0.18 per cent carbon, none of which is present as graphite in the other series. A comparison can thus be made between the changes of volume of the two series under similar tests after fifteen heats, by means of the following:

Alloy	Carbon	Silicon	Per Cent Change in Volume	Alloy	Carbon	Silicon	Per Cent Change in Volume
T.........	0.17	0.65	−0.025	K..........	3.90	0.69	+ 5.40
U.........	0.18	1.10	0.000	N..........	3.98	1.07	+15.20
V.........	0.19	2.71	+0.394	P..........	3.97	2.96	+31.35

19 This comparison serves to emphasize anew that free carbon, even in the form of graphite, is one of the essential factors in the growth of cast irons under heat treatment. The previous series of alloys, however, N to S, brought out clearly the fact that in the constant

graphite series the growth is roughly proportional to silicon present, graphite becoming merely the agent or forming the avenues by means of which the silicon present can be acted upon. It is clear, therefore, that both graphite and silicon are involved in these changes of volume after repeated heatings.

20 A sample of test piece S, known as "S ½," was heated to constant volume in vacuo. This resulted in a shrinkage of 0.04 per cent. The same sample was afterwards heated in the muffle to a constant volume, when a growth of 67.70 per cent was obtained. Fig. 3 shows curves plotted from these data. It will be seen that S ½ grew rapidly during the later heats, with no cracks developed, and the sample retaining its original form throughout. S, however, grew rapidly during the earlier heats, cracks developing during the first heat, and finally breaking in two.

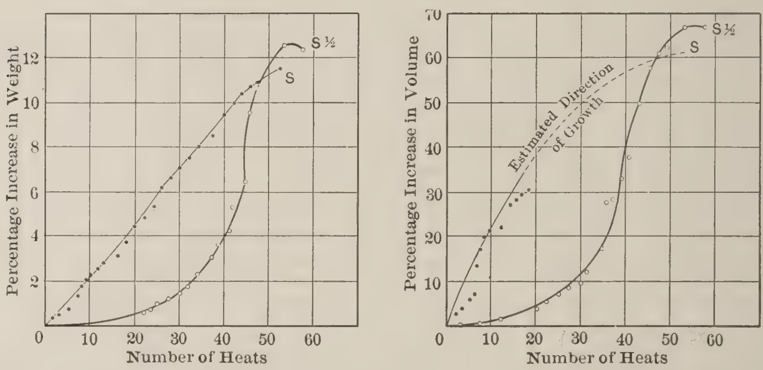


FIG. 3 CURVES SHOWING PERCENTAGE INCREASE IN WEIGHT AND VOLUME AND NUMBER OF HEATS: TEST PIECES *S* AND *S*½

21 Mr. A. Wolfe, superintendent of motive power of the United Railway and Electric Co., Baltimore, commenting upon the growth and final failure of some fittings in one of their power houses says, "The temperature was not constant, varying between that of the temperature of saturated steam at 175 lb. per sq. in. to superheated steam running between 500 deg. fahr. to 550 deg. fahr. total temperature."

22 From the experiments I have made there is considerable evidence indicating that gray cast iron subjected to changing temperatures from 450 deg. fahr. and up gives evidence of an oxidation of the silicon present, forming silica in a micro-crystal form, which upon

cooling causes a disintegration of the surface exposed, ranging generally along the planes formed by the graphite, changing an apparently solid wall into one showing many cracks. It is the constant recurrence of these conditions, produced by the changing temperatures, that, in time not only produces growth, but breaks down the structure of the metal.

23 The experiments conducted by Prof. E. F. Miller were so treated, cooling down each night to the temperature of saturated steam. No mention is made of any growth of these specimens. One would expect to find a marked relation between growth and loss of strength.

24 In a comparison between gray iron samples, those subjected to a heating and cooling treatment totaling 1000 hours would become weaker and larger than those kept constantly at the temperature of superheated steam for a like period. I believe that actual contact with steam is not a necessary condition in this experiment, neither is it a comparative test of the metal, which in service has one surface exposed to atmospheric conditions.

AUTHOR'S CLOSURE

Prof. Ira N. Hollis. The discussion of these papers brings out certain interesting and valuable conclusions which cannot fail to assist in the proper use of cast iron for parts of machinery and boilers. Previously existing differences of experience with this metal under a high temperature are shown to be due to fundamental differences of chemical composition or to variations in the temperature. From this point of view, existing data, even though conflicting, can probably be reconciled. The following conclusions as the result of the papers and discussions may be studied with profit in connection with new construction.

- *a* Cast iron varies in its behavior under high temperature, starting from about 450 deg. fahr. In many cases it deteriorates in structure and strength to a marked degree.
- *b* The effect of high temperature is independent of the medium producing it, whether superheated strain, hot gases or solids.
- *c*. The change of structure or deterioration is much increased by a fluctuating temperature.

2 Where the temperature is constant, even though as high as 600 or 700 deg. fahr., the change in cast iron is not serious enough to pro-

hibit its use, but where the temperature varies considerably, the metal is certain to develop cracks and distortion that render it unsuitable for steam pipes and other parts under steam pressure.

d Cast iron of certain chemical constituents increases materially in volume when subjected to fluctuating temperatures above 500 deg. fahr.

e The chemical composition of the cast iron has a material bearing upon the change of shape and volume and upon the development of imperfections.

3 Certain facts in this connection are well shown by Professor Rugan's experiments. As he states, cast iron containing only combined carbon does not change even under high fluctuating temperatures unless the carbon begins to separate into a graphitic form. Free carbon is one of the factors assisting in the deterioration under high temperature, especially when associated with silicon. The latter seems to be the chief cause of increase in volume. Where the free carbon is constant, the growth in dimensions is roughly proportional to the percentage of silicon.

f The use of cast-iron fittings for superheated strain is inadvisable where the temperature is likely to fluctuate, but it can be safely used where the temperature is to be constant.

g Cast-iron fittings should not be placed in any parts of a steam-pipe line where there are serious bending stresses in addition to the stresses produced by internal pressure, unless the combined stresses are fully allowed for, or neutralized by expansion joints.

GAS POWER SECTION

REPORT OF STANDARDIZATION COMMITTEE

Published in The Journal for December 1909

DISCUSSION PRESENTED AT ANNUAL MEETING 1909

Dr. D. S. Jacobus. I would ask why it is that if the engine, producer and other apparatus, are put in by the same party, it is not necessary that data shall be taken of the efficiency of the producer, as differentiated from the efficiency of the engine, etc. I may have misunderstood that feature of the report, but I remember that there was a severe criticism of a paper presented last year upon a test of a steam power plant, in which simply the fuel and the final output were measured. It was held that what was really needed was the intermediate data to show how the efficiency was obtained.

Prof. R. H. Fernald. In Par. 24*a* of the report it is recommended that the sample be dried at 210 deg. fahr., or just below the atmospheric boiling point. The code of the American Chemical Society requires, I believe, a temperature of 105 deg. cent., or a little above boiling point, which is the Government specification at the present time.

2 One of the most difficult things is to judge the conditions of the producer bed, as the report implies (Par. 25). It is so important that a special bulletin, No. 394, has just been prepared for the technological branch of the geological survey report dealing entirely with the question of the bed. Tests of a 250-h.p. producer carried on for 60 hours continuously, showed that this length of time was not nearly long enough to give conclusive results.

3 In Par. 52 reference is made to the time required to reach the steady state in the case of large engines, which raises a query as to

the division between large and small engines. This depends on individual practice. A 250-h.p. engine might be large from one point of view and small from another.

4 In Par. 55 the committee states, in discussing the speed, "this may be checked by instantaneous readings or intermittent countings taken at regular intervals and numerically averaged." I think that clause should read *must be*. It is almost imperative in any test that we should have a definite check on the readings.

5 In Par. 60 attention is called to "how the cards shall be intergrated," as one of the important considerations. It is difficult to determine exactly the factors which influence the mean effective pressure of the cards. It is easy to have the indicator card so integrated that the brake horsepower will be larger than the indicated horsepower, and the work must be handled with much care.

6 In regard to the characteristics of coal, it might be of interest to have on file a brief outline made for the American Society of Testing Materials. That society intends to draw up specifications on the basis of this outline. It may also be wise to have on record the Test Code adopted by the Government for the plant of the Geological Survey, which though relating to a special plant may assist the committee to get at the general points which have proved of special importance in these investigations.

7 The writer is in sympathy with the committee's recommendation of the use of high actual heat value for the gas, but would like to inquire in regard to liquid fuel. In questions relating to alcohol and gasolene, particularly with alcohol in which the percentage of water is from 15 to 50 per cent, the efficiency curves, which are very beautiful, when worked out on the basis of low heat value, are completely destroyed, and present peculiar characteristics when referred to the high basis. Does the committee intend to include the liquid fuels?

A. A. CARY. I think it is entirely wrong to consider a producer gas plant and its engine as one continuous piece of apparatus, or practically so. In the days of Watt, the engine efficiency was determined by pounds of coal per indicated horse power and what happened between the coal pile and the steam engine indicator was entirely neglected. Finally engineers began to realize that there was a boiler between the coal pile and the engine indicator, which had a part in that efficiency, and we are now careful to get at the efficiency of both boiler and engine. The individual efficiency of both producer and

engine should be determined in producer gas practice, for only in that way can improvements be made.

Edwin D. Dreyfus. In the re-arrangement of the report, from which in its final form such parts as have served merely to guide the committee in choosing and defining the most practical methods of testing will no doubt be eliminated, or incorporated in an appendix, a set of practical and standard tests should be distinctly defined, noting all refinements to be observed and the degree of accuracy to be attained.

2 The question of ratings (Par. 51) might be further elaborated. It is well known that altitude influences the power the engine is capable of developing, by reason of decreased density of mixture and the amount of heat entering the cylinder. Allowance for change in altitude should be specified, roughly about 3.3 per cent per 1000 ft. variation in elevation above sea level or point at which the normal rating is based, which is a fair average for fluctuating atmospheric conditions. Temperatures and pressures are additional factors. In some installations these items might be approximately selected, while in other cases conditions might arise to cause a considerable deviation from normal pressure and temperature. For example, if the supply pipes to the engines are located in a hot basement, and where the engine receives its supply through a holder located at an appreciable distance, maintaining a practically constant pressure, the drop in pressure in the delivery pipe becomes sufficient to cause the engine to pull several inches of suction at maximum load.

3 The question of reponsibility for the expense involved in conducting tests (Par. 87) is a commercial consideration and should not be included in the report. If, however, it is judged contrarily that this expense should be ascribed in some manner to protect the interest of the gas-power industry, the responsibility should then be differentiated in fairness to those builders who have already made large expenditures in proving the efficiency of apparatus, both in the field and in their shops. If small engines are concerned, witness tests may be readily made at the builders' works, which should involve no expense to the purchaser, provided they comply sufficiently with the requirements of the Standard Code of Tests. For large power work, the plant usually has an engineering staff capable of executing the tests as part of their duties, and in this case it would be optional to the builder to have representation when the tests were in progress, or at any time prior to the tests to provide normal adjustment of the apparatus.

4 Confusion in arriving at agreements (Par. 40*b*) and reporting the engine and producer tests separately would be likely to result from the abandonment of the use of the low value and of the distinction between the high and the low values of the gas. The low value has now been so extensively used in engine practice, that it should be clearly defined, and high and low values for different constituents of the gas also included. This will facilitate equating results from one basis to the other, where either one has been arbitrarily used in the past. This characteristic of the gas materially affects its suitability for the gas engine, due to the latent heat of the moisture formed in the burning of the hydrogen gases affecting the temperatures produced by the gas during combustion, and also a high hydrogen content limiting the compression pressures which can be safely used without danger of premature explosions. It should be given due regard, especially where the responsibility for the installation of the engine and producer is divided.

5 The percentage variation stipulated as permissible in the quality of the gas (Par. 40*d*) is more rigid than need be. The engine will work successfully on greater fluctuation, say from 7½ to 10 per cent. Although a good producer will normally produce a quality of gas within these limits, somewhat greater variation might take place for short periods, which though causing no difficulty at the engine, may bring on a disagreement unless this condition is fully appreciated. Some attention should be given to defining distinctly how these measurements should be made.

L. B. Lent. Under general recommendations, Par. 77 reads: "All terms made in a guarantee should be defined. All guaranteed quantities must be capable of measurement and only one acceptable mode of measurement should be specified." Par. 79 reads: "Builders best serve their own interests when units are in terms most satisfactory to the purchaser, and involve only input and output for definite fuel, horsepower and time." If we were to put into a guarantee, which usually goes with the specifications of a gas engine, a description of all the apparatus used, stating the quantities and specifying the modes of measurement, our customers would be apt to entertain a proposition from some one with simpler specifications. If the Society sanctions a code in which the ordinary guarantee must be specified in legal phraseology, protecting the makers' interests until the apparatus is paid for, the guarantee would be far longer than the specifications and rather cumbersome. That part of the

report which deals with recommendations as to guarantees and other relations between manufacturers and builders, should be very carefully considered.

Dr. C. E. Lucke. This discussion will prove helpful to the Committee, and speaking for the Committee I can only hope there will be more of it. The more you consider these questions the more difficult they look. It is a simple matter to rise and recommend that a thing be done so and so, because you have not thought of any other way of doing it, but I suggest that you think about the report for six months, since it is only a preliminary report, and then after you have begun to realize the object of the report, write in your judgment of it.

2 The Committee has been nearly two years at this work and was many times seemingly ready to report, but did not because of the difficulty of the problems. The more views we can get on the questions at issue, the better. I have no intention of defending the report. My object is to ask you to give us your best judgment after mature deliberation, not before, because the questions are questions that are keeping courts busy all over this broad land today. The duty of the Committee was not to devise scientific tests for finding the physical actions involved—that is the business of the Code Committee. The business of this Committee was to eliminate misunderstanding between builders and purchasers, in so far as it could be traced to purely definitional causes from the use of terms having no accepted simple meaning, or from defining quantities not measurable.

3 I would therefore prefer not to reply to each of the suggestions but to let them go on record for the consideration of the Committee; but it is proper that I should say that there is the largest possible difference between tests for scientific information and the tests necessary to define a piece of apparatus sold. It is with the latter kind that this Committee deals and not with the former.

PRELIMINARY REPORT OF LITERATURE COMMITTEE

In accordance with a recommendation from the Chairman of the Meetings Committee of the Section, in a communication published in The Journal for October 1908, a Literature Committee was appointed at a meeting of the Gas Power Executive Committee, January 27, 1909. A tentative program for the work of the committee was outlined by Dr. C. H. Benjamin, *Chairman*, in a verbal report at the Spring Meeting of 1909. It is hoped to index all books, periodicals and transactions, domestic and foreign, dealing with gas power and allied subjects; also to present reviews of new books, and abstracts of important articles. Index cards have been prepared and are being filled by the members of the Committee, the work being divided under the following heads: reviews of French books and periodicals; reviews of German books and periodicals; reviews of Italian books and periodicals; revised list of articles appearing in English and American periodicals; reviews of English and American books as they appear; index of material on file in the library, both books and periodicals. The lists will be published from time to time in The Journal, with an index whenever the accumulation of material warrants.

GAS POWER LITERATURE

ARTICLES IN PERIODICALS[1]

ALCOOL, VINS E BIRRA, E. BOULLANGER. *L'Industria Rivista Tecnica, October 17, 1909.* $1\frac{1}{2}$ pp. *bdfA.*

Deals with the commercial aspects of alcohol power.

AUTOMOBILE-MUFFLER FOR FREE EXHAUST. *Le Genie Civil, September 25, 1909.*

Note abstracting article in La Vie Automobile, July 17, 1909.

CALORIMETRY OF GASES, METHOD AND APPARATUS FOR. *Le Genie Civil, September 11, 1909.* $\frac{1}{4}$ p.

Abstract of note presented by M.M. Lemoult and Jungfleisch to the Academie des Sciences. A simplification based upon loss of volume and consumption of oxygen.

CALORIMETRY IN THE THERMAL LABORATORY OF THE UNIVERSITY OF MOSCOW, METHODS OF, LOUGUININE AND SCHÖNKAREW. *Journal de Chimie Phisique, October 1, 1909.*

Note reviewing the French translation of this Russian work, according to it very high authority. Trans. by Ter Gazarian, Paris: A. Hermann, 1908. 192 pp.

CHEMISTRY SINCE LAVOISIER, HISTORY OF DEVELOPMENT OF. *Revue Universelle des Mines et de la Metallurgy, August 1909.*

Note reviewing French translation of fourth German edition (the first appeared in 1869), by Professor Ladenburgh of the University of Breslau. *Paris*, 1909. P. Hermann.

COMBUSTIBILI. *L'Industria Rivista Tecnica, September 26, 1909.* 1 p. *abdfA.*

From Chemiker Zeitung, 1909, p. 893. Considers mineral oil and gas for illumination, and developments of the oil, gas and coke industry.

EXPLOSIONS, GASEOUS. *Engineering (London), September 3, 1909.* Report to British Association at Winnepeg. $4\frac{1}{2}$ pp., 3 figs., 6 curves. *ceA.*

ENGINES, INFLUENCE OF COMPRESSION PRESSURE UPON THERMAL EFFICIENCIES OF GAS, W. A. TOOKEY. *Engineering (London), October 15, 1909.* 2 pp., 3 curves, 1 table. *cB.*

Discussion of report of Professor Burstall before Institution of Mechanical Engineers.

[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as: *a* comparative; *b* descriptive; *c* experimental; *d* historical; *e* mathematical; *f* practical. A rating is occasionally given by the reviewer, as *A*, *B*, or *C*.

Engine, Variable-Stroke Petrol. *The Engineer (London), October 15, 1909.* ¾ p., 3 figs., 1 table. *bcB.*

A new engine for motor cars giving variable speed on direct drive.

Engine, 600-B.H.P. "Premier" Gas. *Engineering (London), November 19, 1909.* ⅓ page, 1 fig., *bC.*

Three-cylinder, direct-connected, gas-engine; water-cooled piston, engine mechanically oiled, etc.

Engines, Low-Tension Ignition Gear for Gas. *Engineering (London), October 29, 1909.* 1 p., 7 figs., *bC.*

Built for large engines; uses direct current from 65 to 220 volts. Intended to prevent possibility of pre-ignition.

Engine, New Reversible Marine Oil. *The Engineer (London), December 10, 1909.* ¾ page, 3 figs., *bC.*

The "Bolinders" engine, by Jas. Pollock Sons & Co. Two-cylinder, two-cycle, hot-bulb, oil-injected, with reversing gear.

Fuel for Automobiles, Benzol as a Cheap. *Le Genie Civil, October 2, 1909.*

Note abstracting tests reported in L'Automotor, June 26, 1909.

Motors, Balancing Automobile, *Le Genie Civil, September 25, 1909.*

Note abstracting article in La Technique Automobile, July 15, 1909.

Motors for Dirigibles and Aeroplanes, Light Explosion, Ch. Dantin *Le Genie Civil, June 5, 12, 19, 1909.* 18 pp., many figs. *bA.*

This article has not been inspected fully; but its great importance, as a résumé of French practice in this field, makes its inclusion inevitable.

Motors, Safety Crank for Hoists or Explosion. *Bulletin de la Société Industrielle de l'Est, July 1909.* 1 p.

Program for a competition for designs opened by the Association des Industriels de France contre les Accidents du Travail, 4 Blvd. St. André, Paris.

Motori a Combustione Interna—Motori Diesel Reversibili. Ling R. Maraver. *L'Industria Rivista Tecnica, August 15, 1909.* 2 pp., 2 figs., *bcfA.*

Discussion of Diesel 3-cylinder gas engine.

Motori a Combustione Interna. *L'Industria Rivista Tecnica, October 17 1909.* 2 cols., 1 fig., 1 table. *abcA.*

Shows complete sections of producer plant with engine of A. G. Dresdner Gas Motoren-Fabrik gia Moritz Hilledi Dresda.

Motore ad Olio Pesante. *L'Industria Rivista Tecnica, August 8, 1909.* 1½ pp., 2 figs., *bcfB.*

Discussion of gas engine of 3 to 15 h.p.

Motori a Gas e Motoria Vapori. *L'Electricista, May 15, 1909.* 2 pp. *afB.*

Discussion of advantages of gas for power plant economy.

PUMP, AN INTERNAL-COMBUSTION, Herbert A. Humphrey. *Engineering (London), November 26, December 3, 10, 1909.* 13 pp. 17 figs., 13 curves. Also *The Engineer* (*London*), same dates. *bcfA.*

Paper before Institution of Mechanical Engineers. A new principle in pumping water and compressing air. The gas is exploded on the surface of the water in an explosion chamber in one end of a U-tube, the explosion forcing the water up in the other leg. No piston, connecting rod or crank, is used. Since the expansion stroke is longer than the compression, the gas can expand to atmospheric pressure.

PUMP, THE HUMPHREY GAS, W. Cawthorne Unwin. *Engineering* (*London*), *October 15, 1909.* 3½ pp. 8 figs., 4 tables. *bcA.*

Report of tests on Humphrey internal-combustion pump, with general conclusions.

BOOKS

ENGINE, THE GAS. F. R. Hutton. *Wiley, 1908.* 3d. ed. 562 pp., 5¾ x 9, 259 figs. $5.

A treatise on the theory, construction and operation of internal-combustion motors.

ENGINE, THE GAS. Forrest R. Jones. *Wiley, 1909.* 447 pp., 5¾ x 9¼, 142 figs., 30 tables. $4.

Construction, operation, theory and performance of gas engines. Particularly good are the parts dealing with operation, ignition, adjustment and the location of troubles. Useful and practical for the operator as well as for the student.

ENGINE, THE GAS. Cecil P. Poole. *Hill, 1909.* 97 pp., 6 x 9½, folding tables. $1.

Elementary theory, construction, functions of various parts and operation of gas engines. Practical, useful and concise. A good introductory work of strictly limited scope.

ENGINE IN PRINCIPLE AND PRACTICE, THE GAS. A. H. Goldingham. *Gas Power Publishing Company, 1907.* 195 pp., 6 x 9¼, 107 figs. $1.50.

A good elementary book for non-technical readers. No theory is included.

ENGINE THEORY AND DESIGN, GAS. A. C. Mehrtens. *Wiley, 1909.* 250 pp., 5 x 8¼, over 200 figs. $2.50.

Elementary and superficial book, not reliable in its fundamental parts. Contains descriptive matter, and working drawings of small two-cycle and four-cycle engines.

ENGINE, GAS, PETROL AND OIL, Dugald Clerk. *Wiley, 1909.* v., 1 VII + 380 pp., 8vo., 5 plates, 126 figs. *ce.* $4.

Historical sketch and thermodynamics of gas engines. A very complete study of the phenomena of gaseous explosions in closed vessels, with critical comparative examination of results. Followed by a study of explosion phenomena in actual engines by Clerk's "zig-zag" method—and the deduction therefrom of apparent specific heats of gases. Method developed for determining heat balances based on the "apparent" specific heat, and applied to various gas-engine tests. A valuable contribution to the theory of the gas engine—suitable for advanced students.

ENGINES, INTERNAL-COMBUSTION. R. C. Carpenter, H. Diederichs. *Van Nostrand, 1909.* 2d ed. 597 pp., 6 x 9½, numerous text figs. $5.

A comprehensive work of value, including the results of much German investigation for the first time in English. Four sections, dealing with: *a* theoretical considerations; *b* fuels and the phenomena of combustion; *c* construction and operation; *d* power estimation, tests and costs. A good text or reference book.

ENGINES, INTERNAL-COMBUSTION. Wm. H. Hogle. *McGraw, 1909.* 250 pp., 5½ x 8¾, 106 figs., 21 tables. $3.

Contains description and comparison of cyles; practical details of operation; starting devices; carburetors and vaporizers; producers; fuels and combustion; engine compression; the indicator card; design; governing devices; ignition; engine testing; report of tests. Treatment everywhere is elementary; rules of thumb only, given for design, and those only for small engines. Numerous inaccuracies.

ENGINES, CARBURETING AND COMBUSTION IN ALCOHOL. E. Sorel. *Wiley, 1907.* 269 pp., 5 x 8¼, 26 figs. *bcfA*. $3.

A valuable treatise on the properties of alcohol and on carbureting and explosions, based upon the author's extensive research. Unique in this field.

ENGINE, THE INTERNAL-COMBUSTION. H. E. Wimperis. *Van Nostrand, 1908.* 326 pp., 5½ x 8¾, 114 figs. $3.

Devoted primarily to the development of a new theory of gas engines, based upon variable specific heats of the gases. Much critical discussion of gaseous explosion experiments. Gas producers and engines are described rather briefly, and much information is given as to tests, costs, statistics and general data. Gasolene engines are also dealt with. The book takes up a number of special topics which are subjected to mathemetical analysis; as for example, the inertia effect in indicators, the theory of jet carburetors, etc. It is interesting and suggestive throughout and would be most valuable to advanced students.

MOTORS, OIL. G. Lieckfeld. *Lippincott, 1908.* 272 pp., 6 x 9¼, 306 figs. $4.50.

An English translation of a well-known German book. Deals with petroleum, benzol and alcohol engines. Properties of the fuels; historical development of the motors; construction of the parts; examples of modern construction, giving cost, weight, dimensions; applications for various purposes; erection and operation. The book is entirely descriptive, with no theory and but little explanation of functions of parts. A useful book in a limited field.

POWER, GAS. F. E. Junge. *Hill, 1908.* 548 pp., 6 x 9¼, 8 plates, 145 figs. *abdfA*. $5.

Part I: Evolution of gas power and economic aspects; includes a historical and critical study of the development of the present methods. Part II: Design and construction of large gas engines; with descriptions of the most prominent German types. Part III: Applications of gas power in iron and steel, coal-mining and coke-making industries, and elsewhere: use of low-grade fuels. A suggestive and useful treatment of the use of blast-furnace and coke-oven gases for developing large powers.

OTHER SOCIETIES

AMERICAN SOCIETY OF CIVIL ENGINEERS

At the meeting of the American Society of Civil Engineers on April 6, the New York tunnel extension of the Pennsylvania Railroad was discussed, with papers by B. F. Cresson, Jr., on The Terminal Station: West; and F. Lavis, on The Bergen Hill Tunnels. On April 20, Herbert M. Wilson presented a paper, entitled Federal Investigations of Mine Accidents, Structural Materials and Fuels, and the United States Testing Station at Pittsburg.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The 246th meeting of the American Institute of Electrical Engineers was held in Charlotte, N. C., March 30 to April 1. Papers were presented by A. Milmow, on Electric Drive in Textile Mills; by E. D. Latta, Jr., on Gas Engines in City Railway and Light Service; by A. E. Kennelly, on Modifications of Hering's Laws of Furnace Electrodes; by Carl Hering, on The Proportioning of Electrodes for Furnace Electrodes; by E. E. F. Creighton on Some Demonstrations of Lightning Phenomena; by W. S. Lee, on Economics of Hydroelectric Plants; by L. C. Nicholson, on Protecting Insulators from Lightning and Power Arc Effects on the Lines of the Niagara and Lockport Power Co. Visits were made to various plants in the vicinity, including a tour of the Great Falls and Rocky Creek Stations and a 100,000-volt sub-station of the Southern Power Company.

Through the Industrial Power Committee the Institute coöperated with The American Society of Mechanical Engineers in the meeting of April 12, a report of which is given on an earlier page.

At the regular April monthly meeting, which was postponed to April 15, Dr. Samuel Sheldon, professor of physics and electrical engineering at Brooklyn Polytechnic Institute, presented a paper, Education for Leadership in Electrical Engineering, which was discussed by prominent educators.

Circumstances have made it expedient to change the date of the proposed meeting at San Francisco to May 5–7. This meeting is to be under the auspices of the High-Tension Transmission Committee. Arrangements are being made for tours of inspection to important standby plants and receiving stations, transmission plants and mountain generating stations, during that and the following week. The annual meeting of the Institute will be held in New York, May 17.

On May 27, there will be a special meeting, with papers, under the auspices of the Railway Committee. A list of papers for these meetings will be found in another department.

The annual convention will be held June 27-30 at Jefferson, N. H.

AËRO CLUB OF AMERICA

Headquarters have been opened in the Engineering Societies Building by the Aëro Club of America, and the public was invited to the opening meeting in the Auditorium of the building, Wednesday, April 13. A short address by Cortlandt F. Bishop, president of the club, was followed by an exhibit of moving pictures showing recent leading events in aërial navigation and the possibilities of aërial warfare. Views of the club's new aviation field at Mineola, L. I., including the sheds in which members may house their aëroplanes and several machines now under construction, were also thrown on the screen. Following these exercises, a reception was held in the rooms of the club.

NATIONAL METAL TRADES ASSOCIATION

The twelfth annual convention of the National Metal Trades Association was held at the Hotel Astor, New York, April 13, 14, 1910. On Wednesday, besides the routine reports, William Lodge, chairman of the committee on Industrial Education, offered a report reviewing the work of the Winona Technical Institute at Indianapolis, Ind., to the support of which the Association contributes. The papers presented were: Cincinnati's Continuation School, by Dr. Frank B. Dyer; Employers' Liability Insurance, by Miles Dawson; The Growth of the Coöperating System, by Prof. Herman Schneider; Insurance against Unemployment, by John L. Griffiths. The papers presented at the Thursday morning session were: Modern Methods of Shop Management, by Fred. S. Waldron, Mem.Am.Soc.M.E.; Premium Systems, by Carl G. Barth, Mem.Am.Soc.M.E.; Cincinnati's Continuation School, by J. Howard Renshaw. The annual banquet took place Wednesday evening at the Hotel Astor. President Wells presided as toastmaster, and the diners numbered about two hundred and fifty. J. H. Schwacke was elected president of the association.

ENGINEERS SOCIETY OF WESTERN PENNSYLVANIA

Notice has been received at this office of the removal April 27 of the Engineers Society of Western Pennsylvania to their new quarters in the Sixth Avenue wing, 25th floor, of the Oliver Building, Sixth Avenue and Smithfield Street, Pittsburg, Pa.

PERSONALS

John R. Allen, professor of machanical engineering, University of Michigan, will head the engineering school to be established at Robert College, Constantinople. Professor Allen will go to Turkey early next summer.

A biographical sketch of Prof. L. P. Breckenridge was published in the April issue of *Cassier's Magazine.*

Edward W. Burgess, formerly in the employ of the Metzger Motor Car Company, Detroit, Mich., has become connected with the J. I. Threshing Machine Company, Racine, Wis.

Gordon M. Campbell, formerly superintendent of the power apparatus shops of the Western Electric Co., Chicago, Ill., has become connected with the turbine department of the General Electric Co., West Lynn, Mass.

A. W. Cash, mechanical engineer and expert, recently located at Newark, N. J., has taken charge of the regulating valve and engineering department of the H. Mueller Mfg. Co., Decatur, Ill.

Walter Castanedo has been appointed manager for the New Orleans, La., district of the Harrisburg Foundry and Machine Works. He was formerly a member of the firm of Glenny & Castanedo, New Orleans, La.

S. M. Chandler has been appointed president of the Chandler-Boyd Supply Company, Pittsburg, Pa. He was formerly associated with the Pittsburg Valve and Fittings Company, Barberton, O.

Philip G. Darling, recently mechanical engineer of Manning, Maxwell & Moore, Inc., New York, has become associated with the development department of the E. I. du Pont de Nemours Powder Co., Wilmington, Del.

William W. Estes has accepted a position with the General Fire Extinguisher Company, Providence, R. I., as designer. Mr. Estes was until recently chief engineer of the R. I. Co., of the same city.

A. L. G. Fritz has become chief draftsman of the Hartford Suspension Company, Jersey City, N. J. Mr. Fritz was formerly associated with the Tee Square and Triangle Co., Newark, N. J.

Reuben Hill, formerly factory manager of the Bristol Engineering Corporation, Bristol, Conn., has entered the service of the Hudson Motor Car Company, Detroit, Mich.

Dr. D. S. Jacobus read a paper on Superheated Steam and Superheaters at the annual meeting of the National Association of Cotton Manufacturers, Boston, April 27, 28.

Charles Kirchhoff sailed for Europe April 19, expecting to be abroad for several months. He will sojourn for some time in Italy, and from there go to Germany.

Charles W. Lummis, recently mechanical engineer of the Camden Iron Works, Camden, N. J., has become identified with the Scovill Manufacturing Co., Waterbury, Conn.

Walter M. McFarland, who has occupied the office of acting vice-president for the Westinghouse Electric & Mfg. Co., Pittsburg, Pa., for a period extending over ten years, has resigned to accept an official position with the Babcock & Wilcox Co., Singer Building, New York.

D. H. Macdonald has entered the service of the Southern Engine and Boiler Works, Jackson, Tenn. He was formerly identified with the Minneapolis Steel and Machinery Company, Minneapolis, Minn., as general superintendent of the mechanical department.

John B. Mayo, formerly connected with the Crocker-Wheeler Co., Ampere, N. J., has been appointed mechanical engineer and chief draftsman of the Texas Portland Cement Company, Cement, Texas.

Fred J. Miller has been elected a member of the Board of Directors of the Union Typewriter Company.

Albert B. Moore, formerly chief engineer of the Griffin Wheel Company, Chicago, Ill., has resigned his position and formed a partnership with Andrew W. Woodman, under the firm name of Woodman & Moore, civil and mechanical engineers, with offices in the People's Gas Building, Chicago, Ill.

Francis P. Ritchie has accepted the position of factory manager of the Berliner Gramophone Company, Montreal, Canada. He was recently superintendent of the Northern Electric and Mfg. Co., of the same city.

William J. Sando, manager of the pumping engine and hydraulic turbine department of the Allis-Chalmers Co., for nearly six years, has resigned. After taking a few months' rest Mr. Sando will open an office in Boston as consulting engineer.

Walter G. Scott, recently associated with the Allis-Chalmers Co., West Allis, Wis., has become identified with the Cyclone Drill Company, Orville, O.

John J. Swan has become connected with the Keller Mfg. Co., of Philadelphia, Pa. Until recently he was secretary of the W. P. Pressinger Co., New York.

Charles E. Sweet, formerly superintendent of steam turbine construction, Westinghouse Machine Company, East Pittsburg, Pa., has become associated with the E. M. F. Co., Detroit, Mich.

Charles W. Werst has accepted a position with the Lima Locomotive and Machine Co., Lima, O., in the capacity of assistant superintendent. He was until recently general foreman of the Baldwin Locomotive Works, Philadelphia, Pa.

CURRENT BOOKS

American Machinists' Handbook and Dictionary of Shop Terms. A Reference Book of Machine Shop and Drawing Room Data, Methods and Definitions. By Fred H. Colvin and Frank A. Stanley. *New York: McGraw-Hill Book Co., 1909.* Morocco, pocket-book size, xxi +513 pp., illustrated. Price, $3.

Contents: Screw Threads: Cutting Screw Threads, Standard Proportions of Screw Threads, Measuring Screw Threads; Pipe and Pipe Threads; Twist Drills and Taps; Taps; Files; Work Benches; Soldering; Gearing; Milling and Milling Cutters; Cam Milling Machine Feeds and Speeds, Tables for Use with the Dividing Head, Milling Cutter, Reamer and Tap Flutes; Grinding and Lapping; Grinding Wheels and Grinding, Lapping, Reamer and Cutter Grinding; Screw Machine Tools, Speeds and Feeds: Types of Tools and their Construction, Speeds and Feeds for Screw Machine Work; Punch Press Tools; Bolts, Nuts and Screws: Tables of Cap and Machine Screw Dimensions, Tables of A. S. M.E. Standard Machine Screw Dimensions, Nut and Bolt Tables, Miscellaneous Tables; Calipering and Fitting: Press and Running Fits, Dimensions of Keys and Key-Seats; Tapers and Dovetails: Measuring Tapers, Tables of Standard Tapers; Shop and Drawing Room Standards: Standard Jig Parts, Tables of Dimensions of Standard Machine Parts, Miscellaneous Tables; Wire Gages and Stock Weights; Belts and Shafting; Steel and Other Metals; General Reference Tables; Shop Trigonometry; Dictionary of Shop Terms.

The American Practice of Gas Piping and Gas Lighting in Buildings. By Wm. Paul Gebhard. *New York, McGraw-Hill Book Co., 1908.* Cloth, 8vo, 306 pp. Price, $3.

Contents: Prejudices against the Use of Gas; Popular Fallacies about Gas; Advantages of Gas as an Illuminant; Advantages of Gas as a Source of Heat and Power; The Arrangement of Gas Piping in Buildings; Specification for Gas Piping for Coal or Water Gas; Rules, Tables and Regulations of Gas Companies and of Building Departments; Piping for Natural Gas; Piping for Air Gas or Gasolene Machine Gas; Piping for Acetylene Gas; The Testing of Gas Pipes; Gas-Light Illumination; Gas Burners; Gas-Pressure Regulation; Gas Globes and Globe Holders; Gas Fixtures; Gas Meters and Gas Meter Stories; The Illumination of Interiors with Gas Lights; The Lighting of Country Houses; The Relations between Gas Companies and Gas Consumers; Practical Hints for Gas Consumers; Some Facts about the Gas Supply; Accidents with Gas; Dangers to the Public Health from Illuminating and Fuel Gas; Historical Notes on the Development and Progress of the Gas Industry; Bibliography of Gas Lighting.

The Design and Construction of Oil Engines, with full directions for erecting, testing, installing, running and repairing; including descriptions of American and English kerosene oil engines. By A. H. Goldingham. 3d edition, revised and enlarged. *New York, Spon & Chamberlain, 1910.* Cloth, 12mo, viii+260 pp., illustrated. Price, $2.50.

Contents: Introductory; On Designing Oil Engines; Testing Engines; Cooling Water Tanks and Other Details; Oil Engines Driving Dynamos; Oil Engines Connected to Air Compressors, Water-Pumps, etc.; Instructions for Running Oil Engines; Repairs; Oil Engine Troubles; Various Engines Described; Portable Engines; Large Sized Engines; Fuels; Miscellaneous.

PRACTICE AND THEORY OF THE INJECTOR. By Strickland L. Kneass. 3d edition revised and enlarged. *New York, John Wiley & Sons, 1910.* Cloth, 8vo, 175 pp., illustrated. Price, $1.50.

Contents: Early History; Development of the Principle; Definition of Terms-Description of the Important Parts of the Injector; The Delivery Tube; The Combining Tube; The Steam Nozzle; The Action of the Injector; Application of the Injector—American and Foreign Practice;Determination of Size—Tests; Requirements of Modern Railroad Practice—Repairs—Methods of Feeding Locomotive Boilers; Feed Water Heating—Efficient Feeding—Flue Mileage—Scale-Bearing Water—Check Valves.

PUMPING ENGINES FOR WATER WORKS. By Charles Arthur Hague. *New York, McGraw-Hill Book Co., 1907.* Cloth, 8vo, xi + 372 pp., illustrated. Price, $5.

Contents: The Pumping Engine; Historical; Economic Steam Duty; The Advent of Triple Expansion; The Mariotte Curve; Steam Jackets; Coal Duty of Pumping Engines; Actual Conditions of Pumping; The Worthington Duplex Pumping Engine; The Gaskill Pumping Engine; The Reynolds Triple Expansion Pumping Engine; Various Types and Classes; Pumping Engines adapted to Conditions; Installation of Pumping Engines; Investment Value of Pumping Engines; Suction Lift and Suction Pipes; Water Passages and Water Valves; The Water Plungers; Air Chambers; Steam Piston; Steam Cylinders; Cross Heads; Frames and Bedplates; Material for Pumping Engines; Duty Tests of Pumping Engines.

DRAWINGS FOR MEDIUM SIZED REPETITION WORK. With Examples for Motor Car parts. By R. D. Spinney. *London, E. & F. N. Spon, Ltd., 1909.* Cloth 8 vo, + 91 pp., illustrated. Price. $1.50.

Contents: Drawings Generally; Standard Lists; Indexes; Tolerances; Dimensioning; Notes on Designing for Repetition Work; Drawing Office Routine.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M.E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

American Machinists' Handbook and Dictionary of Shop Terms. By F. H. Colvin and F. A. Stanley. *New York, McGraw-Hill Book Company, 1909.*

American Practice of Gas Piping and Gas Lighting in Buildings. By W. P. Gerhard. *New York, McGraw-Hill Book Company, 1908.*

American Railway Association. Statistical bulletins Nos. 66, 67-B. *Chicago, 1910.*

The American Society of Mechanical Engineers. The Journal, vol. 31, Nos. 1-12. 2 vols. *New York, 1909.*

Ballistic Electro-Dynamometer Method of Measuring Hysteresis Loss in Iron. University of Kansas, Engineering Experiment Station, Bulletin No. 1. By M. E. Rice and Burton McCollum. *Lawrence, 1909.* (Gift.)

Briquetted Coal and its Value as a Railroad Fuel. By C. T. Malcolmson.

Design and Construction of Oil Engines. Ed. 3. By A. H. Goldingham. *New York, Spon & Chamberlain, 1910.*

Design of Turbo Field Magnets for Alternate-Current Generators, with Special Reference to Large Units at High Speeds. By Miles Walker. Institution of Electrical Engineers, 1909. (Gift of Calvin W. Rice.)

Detroit Board of Water Commissioners. 57th annual report. *Detroit, 1909.* (Gift.)

Drawings for Medium-Sized Repetition Work. By R. D. Spinney. *New York, Spon & Chamberlain, 1909.*

Electric Railway and Lighting Properties. By Stone & Webster. *Boston, 1910.* (Gift of Calvin W. Rice.)

Epitome of the Work of the Aeronautic Society from July 1908 to December 1909. (Bulletin no. 1) *New York, 1909.* (Gift of Aeronautic Society, New York.)

Fifth Avenue Building Site of Fifth Ave. Hotel, Madison Square, New York. *New York.* (Gift of Fifth Avenue Building Company.)

Geology of the Auburn–Genoa Quadrangles. (New York State Museum, Bulletin no. 137.) By D. D. Luther. *Albany, 1910.* (Gift.)

Hawaii. Government and conditions before and since annexation to the United States and present requirements. Speech of Chauncey M. Depew, February 24, 1910. *Washington, 1910.* (Gift).

Innoko Gold–Placer District, Alaska. (Bulletin no. 410, U. S. Geological Survey.) By A. G. Maddren. *Washington, 1910.* (Gift.)

Iron Ores, Fuels and Fluxes of the Birmingham District, Alaba ma (Bulletin no. 400, U. S. Geological Survey.) By E. F. Burchard and Charles Butts. *Washington, 1910.* (Gift.)

Legal Aid Society. 24th annual report of the president, treasurer and attorneys. *New York, 1909.* (Gift.)

Locomotive Coaling Stations. (Bulletin no. 15, Roberts & Schaefer Co., Chicago.) *Chicago.* (Gift of author.)

Modern Coal Mining Plants and Washeries. (Bulletin no. 17, Roberts & Schaefer Co., Chicago.) *Chicago.* (Gift of author.)

Paleontology of the Coalinga District, Fresno and Kings Counties. California. (Bulletin no. 396, U. S. Geological Survey.) By Ralph Arnold. *Washington, 1909.* (Gift.)

Planning and Building of Industrial Plants. Section II. By Charles Day. *Dodge & Day. Philadelphia, 1909.*
Typical Operations. No. 2.

Possibilities of Electrical Power Transmission for Main Propulsions and Speed Regulation. By W. P. Durtnall. (Gift of author.)

Practice and Theory of the Injector. Ed. 3. By S. L. Kneass. *New York, J. Wiley & Sons, 1910.*

Progress of Electrical Braking on the Glasgow Corporation Tramway System. Institution of Electrical Engineers, 1910. By A. Gerrard. (Gift of Calvin W. Rice.)

Reconnaissance of some Mining Camps in Elko, Lander and Eureka Counties, Nevada. (Bulletin no. 408, U. S. Geological Survey.) By W. H. Emmons. *Washington, 1910.* (Gift.)

Results of Spirit–Leveling in Illinois, 1896 to 1908, inclusive. (Bulletin no. 421, U. S. Geological Survey.) *Washington, 1910.* (Gift.)

Short–Circuiting of Large Electric Generators and the Resulting Forces on Armature Windings. By Miles Walker. Institution of Electrical Engineers, 1909. (Gift of Calvin W. Rice.)

Tables of Efficiencies in Percentages, Butt and Double Strap Joints. Computed by W. W. Ramsay. *Boston 1910.* (Gift of Commonwealth of Massachusetts Board of Boiler Rules.)

TESTS OF VERTICAL PUMPING AT QUACKENBUSH PUMPING STATION, ALBANY N. Y. *Chicago, 1909.* (Gift of H. A. Allen & Co.)

TRINITY'S TENEMENTS. CONDENSED REPORT. *New York, 1909.* (Gift of Corporation of Trinity Church.)

EXCHANGES

AMERICAN GAS INSTITUTE. Directory of Membership, 1909. *Easton, 1909.*

AMERICAN SOCIETY OF CIVIL ENGINEERS. Constitution and List of Members, February 1910. *New York, 1910.*

AMERICAN SOCIETY OF CIVIL ENGINEERS. Transactions, vol. 66. *New York, 1910.*

AMERICAN STREET AND INTERURBAN RAILWAY ASSOCIATION. Membership list, 1910. *New York, 1910.*

INSTITUTION OF CIVIL ENGINEERS. Minutes of proceedings, vol. 179. *London, 1910.*

INSTITUTION OF ENGINEERS AND SHIPBUILDERS IN SCOTLAND. Transactions vol. 52. *Glasgow, 1909.*

MACHINERY MARKET. March 4, 1910-date *London, 1910-date.*

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS. Transactions, no. 87. *Boston, 1910.*

NEW YORK STATE MUSEUM. 62d annual report, vol. 1, 1908. *Albany, 1909.*

ROYAL SOCIETY OF NEW SOUTH WALES. Journal, vol. 42, 43, pt. 1. *Sydney, 1908-1909.*

THERMAL CONDUCTIVITY OF FIRE-CLAY AT HIGH TEMPERATURES. University of Illinois, Engineering Experiment Station, Bulletin no. 36. By J. K. Clement and W. L. Egy. *Urbana, 1909.*

UNITED ENGINEERING SOCIETY

INTERNATIONAL WHO'S WHO, 1910–1911. *New York, International Who's Who Publishing Company, 1910.*

DESIGNING AND DETAILING OF SIMPLE STEEL STRUCTURES. By C. T. Morris. *Columbus, 1909.*

GIFT OF SCHNEIDER & CO.

Acier au Manganese, propriétés et applications. Extract from Le Génie Civil. *Paris, 1908.*

Acier au Manganese. *1909.*

Acier Speciaux pour Pièces d'Automobiles. Objets exposés. *Paris, 1906.*

Album des Fers et Aciers. *1895.*

Etablissements de MM. Schneider & Cie. *Nevers, 1900.*
Materiel automobile. *1909.*
Materiel Electrique à Courants Alternatifs. Alternateurs. *1909.*
——— Dynamos Schneider Type "*E*" 1909, type "*S*" 1909.
——— Moteurs triphases, 1905.
——— Transformateurs, 1907.

Principales Installations Comportant des Dynamos "Schneider" a Courant Continu. Liste des references, 1905.

Annexe no 1. *1906-1907.*

Principales Installations Comportant du Material Electrique à Courants Alternatifs de notre Dernier Type. Liste des references, 1906.

Annexe no. 1. *1907.*

Travaux d'amelioration du Port du Havre. *Paris, 1907.*

TRADE CATALOGUES

Avery Scale Company, *North Milwaukee, Wis.* Automatic coal weighing, 24 pp.
Blake & Knowles Steam Pump Works, *New York.* The Blake-Knowles open feed water heaters, 12 pp.

Compagnie des Forges et Aciéries de la Marine et d'Homecourt. *France.* Iron and steel parts for automobiles for buildings, castings of all kinds, wheels, projectiles, guns, fire arms. *75pp.*

G. Drouvé Co., *Bridgeport, Conn.* The anti-pluvius puttyless skylight, and the Lovell window operator, 6 pp.

General Electric Company, *Schenectady, N. Y.* Price list No. 5214: G. E. Edison, carbon incandescent lamps, 7 pp; No. 5218: G. E. train-lighting lamps, 3pp.; Mazda vs. Tungsten lamps, 6 pp.; Bulletin No. 4716: Thomson watt-hour meters with prepayment attachments for direct and alternating currents, 7 pp.; No. 4719, G. E. fan motors and small power motors, 36 pp.

Goldschmidt-Thermit Company, *New York.* Reactions—first quarter, a quarterly periodical devoted to the science of aluminothermics, 20 pp.

Greene, Ketchum & Co., *New York.* Sullivan smokeless furnace equipment, 16 pp.

H. G. Hammett, *Troy, N. Y.* Trojan metallic packing for locomotive piston rods and valve-stems, 8 pp.; Richardson locomotive valves, 15 pp.; Trojan bell ringer for locomotives, 2 pp.; Samson bell ringer for locomotives, 2 pp.; Link grinders, 2 pp.; Radias grinder, 2 pp.; Triple-valve bushing roller, 2 pp.

Ironworks Company, *Jersey City, N. J.* Crowe mechanical stokers for stationary, marine or locomotive boilers, 36 pp.

Andrew J. Morse & Son, *Boston, Mass.* Illustrated catalogue of diving apparatus and other submarine appliances, 72 pp.

Nason Mfg. Co., *New York.* Nason "Vesuvius" steam trap, 8 pp.; Class B, Class C, and sidelug steam traps, 7 pp.

National Lock Washer Company, *Newark, N. J.* Catalogue of car curtains, curtain fixtures, sash locks, sash balances, nut locks, etc., 40 pp.

Nazel Engineering and Machine Works, *Philadelphia, Pa.* Béché patent pneumatic power hammer, 12 pp.

New Jersey Car Spring and Rubber Company, *Jersey City, N. J.* Rubber goods, as fire hose, packing, belts, mats, gaskets, etc., 132 pp.

New York Continental Jewel Filtration Co., *New York.* Mechanical filtration, water purification, filtration of public water supplies, 64 pp.

North Western Expanded Metal Co., *Chicago, Ill.* Expanded metal in use on highway bridges, 16 pp.; Designing data for the use of expanded metal and expanded metal lath, 70 pp.

Reagan Grate Bar Co., *Philadelphia, Pa.* Improved chopping grates for stationary boilers, 40 pp.; Improved chopping grates for marine and locomotive boilers, 28 pp.

Roberts Filter Mfg. Co., *Philadelphia, Pa.* Pressure water filters and filter appurtenances, 72 pp.; Improved water filter for domestic use, 12 pp.

Roberts & Schaefer Co., *Chicago.* Bulletin No. 17: Construction of coal-mining plants, coal-storage plants, mine-ventilating plants, in U. S., 116 pp.; Bulletin No. 15: Construction work of the railroad department, during 1905–1907, 75 pp.

Templeton Mfg. Co., *Boston, Mass.* Sterling steam trap, 7 pp.

Under-Feed Stoker Company of America, *Chicago.* Publicity Magazine, March 1910, devoted to the interests of the Jones stoker, 16 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

022 Engineer experienced in the design and construction of single and multistage centrifugal and turbo-pumps. Must be thoroughly competent to make estimates and prepare complete calculations and data for drafting room. Location, Middle West.

023 Chief draftsman on hydraulic-turbine machinery and hydro-electric plants; one capable of handling draftsmen and pushing work through the drafting room in a systematic and economical manner. Location, Middle West.

024 Instructor in mechanical and architectural drawing, Tuesday and Thursday evenings, October to May. Drafting room practice and some experience in teaching necessary. Location, one hour from Manhattan.

025 Glass company, Western Pennsylvania, requires technical man for general engineering work covering testing of furnaces, boilers and producers; advice on purchase of materials and supplies; design and supervision of new construction. A good working knowledge of chemistry and several years experience in similar work desirable.

MEN AVAILABLE

58 Member, Cornell graduate; twelve years varied experience electrical construction, steam and electrical power-plant equipment, operation and repairs, ice-making and refrigeration engineering and construction; desires position as chief engineer or master mechanic of large industrial or ice and cold storage plant.

59 Member, fifteen years experience in design and construction; seven years teaching; now professor of Mechanical Engineering in the South, wants similar position in the North or Central West or with an established engineering concern.

60 Member, graduate in mechanical engineering with degrees of B.S. and M.E.; two years practical experience in machine design and construction; five years as instructor in mechanical engineering in State university; desires position as assistant professor of mechanical engineering or of machine design. Location immaterial.

61 Member of the Society, graduate engineer, age 57, fifteen years practical experience, drafting room, foundry and machine shop; selling machinery; desires position as general superintendent or manager, where interest in profitable business can be acquired. At present employed.

62 Associate member, age 33, five years practical experience in manufacturing, now head of mechanical engineering department of a Southern technical college, desires position for the summer months with some good manufacturing firm. Would prefer to act as representative in some part of the South, but will consider work of other kinds. Specially qualified along the lines of gas and steam engines and of refrigerating machinery.

63 Member, wants position as general superintendent or works manager; age 35, practical mechanic, technical education, good hustler and organizer; now superintendent of plant doing heaviest class of iron and steel work. At liberty June first, $5000 a year.

64 Junior, technical graduate, several years experience in shop, office, drafting room and testing, with unusual executive ability, force and adaptability, desires to make change.

65 Associate, technical graduate, two years mechanical and electrical draftsman, five years experience in teaching and engineering research, at present engaged in teaching, desires a change. Would like position as head of department of mechanical engineering in some small technical school offering possibilities of growth and development, or as professor or assistant professor of experimental engineering in some large college desiring to develop laboratory and research work. Salary expected $2200.

66 Responsible position in a prominent university desired by engineer; for the past nine years head of a college of Mechanical and Electrical Engineering. Good reasons for changing, record open for inspection.

67 Member, engineer graduate of U. S. Naval Academy, large and influential engineering acquaintance and broad experience in correspondence, selling, and manufacturing in large and well-known works, holding executive positions of responsibility, desires a position of trust with a good manufacturing concern, or responsible position on the commercial end of a business, as branch manager in Boston or elsewhere.

68 Member, chief draftsman in or near New York City. Experience in steam pumps, air compressors, condensers, Corliss engines.

69 Member, graduate M.E. and C.E., twenty-three years experience in general engineering, ten in hydroelectric developments. Charge of the design and installation of power plants in this and other countries; desires position, East preferred.

70 Member desires to communicate with concerns preferably in Pennsylvania, established in general powerplant engineering and contracting work, with a view of buying an interest and assuming an active part in the business; or taking an interest in some manufacturing concern building power-plant equipment or accessories.

71 Technical graduate, experienced in several varied lines of industry, holding executive positions of responsibility during the last eight or nine years, desires to become associated in position of trust with good manufacturing concern, preferably located in the East or Middle West. Best of references.

72 Sales manager, general manager or assistant with manufacturer of power plant apparatus, "water-tube" boiler manufacturer preferred; Junior member, age 32. References can be furnished that will be satisfactory to the most critical. Preferred location, Philadelphia or New York.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

BOCORSELSKI, F. E. (1907), Asst. Mech. Supt., Am. Loco. Co., and 1843 W. Grace St., Richmond, Va.

BRYAN, William H. (1891), Cons. Mech. and Elec. Engr., 315 and 316 Title Guaranty Bldg., and 5718 Vernon Ave., St. Louis, Mo.

BURGESS, Edward W. (Junior, 1908), J. I. Case Threshing Mch. Co., Racine, Wis.

BURGOON, Charles Eli (1907), Ch. Engr., U. S. Court House and P. O. Bldg., and 3824 Rokeby St., Chicago, Ill.

CARLE, Nathaniel A. (1907), Cons. and Contr. Engr., 510 Central Bldg., Seattle, Wash.

CARPENTER, Allan O. (Associate, 1909), Granville Center, Bradford Co., Pa.

CASTANEDO, Walter (1907; 1909), Dist. Mgr., Harrisburg Fdy. & Mch. Wks., 1103 Hennen Bldg., and *for mail*, 1514 Peters Ave., New Orleans, La.

CHANDLER, Sellers McKee (Junior, 1905), Pres., Chandler-Boyd Supply Co., Terminal Warehouses, and *for mail*, 741 Browne St., Shadyside, Pittsburg, Pa.

DARLING, Philip G. (Associate, 1906), Development Dept., E. I. du Pont de Nemours Powder Co., Wilmington, Del.

ENGLISH, Harry K. (Associate, 1908), Genl. Elec. Co., Monadnock Bldg., Chicago, Ill.

FRITZ, Aime L. G. (Junior, 1907), Ch. Draftsman, Hartford Suspension Co., 150 Bay St., Jersey City, N. J.

GERRISH, William H. (1901), 15 Gould Ave., Malden, Mass.

GORDON, Frederic W. (1880), Clardon, Orchard Lane, Ft. Washington, Pa.

GUMP, Walter B. (Junior, 1902), Mech. and Elec. Engr., 206 Kerckhoff Bldg., Los Angeles, Cal.

HAGY, J. L. (Junior, 1901), Harrison Safety Boiler Wks., Tioga, and *for mail*, 1525 Montgomery Ave., Philadelphia, Pa.

HENES, Harry Wm. (Junior, 1909), 557 Barry Ave., Chicago, Ill.

HENSHAW, Frederick V. (1900), Cons. Engr., 24 Broad St., New York, and 79 State St., Brooklyn, N. Y.

HILL, Reuben (1908), Hudson Motor Car Co., Detroit, Mich.

JONES, Charles E. (1894), Mech. Engr., Instr. in Forging, Manual Training Sch. of Washington Univ., 5361 Von Verson Ave., St. Louis, Mo.

KEAN, A. J. A. (1908), Ch. Operating Engr., Guanajuato Power & Elec. Co., Apartado 50, Guanajuato, Mexico.

KENYON, Alfred Lewis (1904), care Dr. A. C. Griggs, Warrensburg, Mo.

KREUTZBERG, Otto August (Associate, 1904), V. P., Pfannmueller Engrg. Co., 3701 S. Ashland Ave., and 38 Roslyn Pl., Chicago, Ill.

LAMBIE, James M. (Junior, 1908), Asst. Genl. Mgr., Findlay Clay Pot Co., Findlay, O., and *for mail*, 306 E. Maiden St., Washington, Pa.

LUCAS, Henry M. (1904), Lucas Mch. Tool Co., E. 99th St. and L. S. & M. S. R. R., Cleveland, and *for mail*, 82 Page Ave., East Cleveland, O.

LUMMIS, Charles W. (Associate, 1907), Scovill Mfg. Co., Waterbury, Conn.

MACDONALD, Duncan H. (1903), Southern Eng. & Boiler Wks., Jackson, Tenn.

MASON, William (1880), 78 Burncoat St., Worcester, Mass.

MAYO, John B. (1892;1894), Mech. Engr. and Ch. Draftsman, Texas Portland Cement Co., Cement, Tex.

MERRITT, Joseph (1907), Cons. and Constr. Mech. Engr., 60 Prospect St., and 64 Deerfield Ave., Hartford, Conn.

MOORE, Albert B. (1903), Member of Firm, Woodman & Moore, Civ. and Mech. Engrs., Peoples Gas Bldg., and *for mail*, 5426 Jefferson Ave., Chicago, Ill.

NORRIS, William H., Jr. (Junior, 1909), Engr., W. R. Grace & Co., Lima, Peru.

OLIVER, E. C. (Junior, 1902), 475 Second Ave., Detroit, Mich.

OWEN, Ira June (Junior, 1905), Cons. Engr., Rm. 502, 115 Dearborn St., Chicago, and 110 Maple Ave., Oak Park, Ill.

PARK, Walter E. (1903), Young & Park, 45 Broadway, New York, N. Y.

PITKIN, Joseph Lovell (Associate, 1903), 72 Washington St., Atlanta, Ga.

PRESSINGER, W. P. (Associate, 1903), V. P. and Mgr. of Sales, Keller Mfg. Co., 21st St. and Allegheny Ave., Philadelphia, Pa.

RIGDON, Carl (Junior, 1907), 710 Market St., Chattanooga, Tenn.

RITCHIE, Francis P. (Associate, 1908), Factory Mgr., Berliner Gramophone Co., and *for mail*, 32 Sussex Ave., Montreal, Canada.

ROGERS, Charles Edward (Associate, 1898), Genl. Mgr., Fraser & Chalmers, Ltd., Corner House, and The Pines, Gordon Hill Rd., Parktown, Johannesburg, South Africa.

ROGERS, Robert W. (Junior, 1908), Mech. Engr., C. A. Stickney Co., and *for mail*, The Meadows, Dodd Pl., St. Paul, Minn.

RYDER, Malcolm P. (Associate, 1901), Engr., Witherbee Igniter Co., Springfield, Mass., and 11 Warren St., White Plains, N. Y.

SHALLENBERGER, Louis R. (1895; 1902), Box 55, Fenton, Mich.

STIMSON, Oscar M. (1906), O. M. Stimson & Co., First Natl. Bank Bldg., Chicago, Ill.

STREET, Clement F. (1893), with firm of Clement F. Street, Locomotive Stokers, 1427 Schofield Bldg., Cleveland, O.

SWAN, John Joseph (1899; 1909), Keller Mfg. Co., 21st and Lippincott Sts., Philadelphia, Pa., and Plainfield, N. J.

SWEET, Charles E. (1907), E. M. F. Co., Detroit, Mich.

SWENSON, Bernard Victor (1903), Barron G. Collier, Inc., 175 Fifth Ave., New York, N. Y.

THOMAS, Fred H. (Junior, 1909), Sales Engr., C. & G. Cooper Co., Mt. Vernon, O.

WALKER, Frank A. (Junior, 1909), with B. B. & R. Knight, and *for mail*, 46 Ring St., Providence, R. I.

WERST, Chas. Wm. (1909), Asst. Supt., Lima Loco. & Mch. Co., and *for mail*, 1160 W. High St., Lima, O.

WILLIAMS, Alan Gillespie (Junior, 1909), 112 Central Ave., South Oil City, Pa.

WOODS, Samuel Hamilton (Junior, 1907), Richardson & Boynton Co., 31 W. 31st St., and 73 W. 12th St., New York, N. Y.

YOUNG, William A. (1901; 1905; 1906), Ch. Draftsman, Morgan Engrg. Co., and *for mail*, 522 S. Arch St., Alliance, O.

NEW MEMBERS

ARNOTT, R. Fleming (Associate, 1909), Cons. Engr., 95-97 Liberty St., New York, N. Y.

BECK, Rudolph H. (Junior, 1909), United Iron Works, Seattle, Wash.

BONNEY, Herbert Marshall (Junior, 1909), Ch. Draftsman, L. F. Fales, and *for mail*, Walpole, Mass.

NELSON, Eric Hugo (1909), Ch. Draftsman, Geo. F. Blake Mfg. Co., Cambridge, and *for mail*, 61 Leverett Ave., Beachmont, Mass.

DEATHS

BLESSING, James H., February 21, 1910.
BRIDGE, James W., December 20, 1909.
CHURCHILL, William W., March 25, 1910.
HASKINS, Harry S., March 13, 1910.
REDWOOD, Iltyd I., April 5, 1910.
SIMS, Gardiner C., March 19, 1910.
SIRICH, J. Henry, Jr., January 22, 1910.

GAS POWER SECTION

CHANGES OF ADDRESS

ENGLISH, Harry K. (1909), Mem. Am. Soc. M. E.

FISKE, Geo. Wallace (Affiliate, 1909), 816 Huntoon St., Topeka, Kan.

HOPCROFT, Ernest Bigly (Affiliate, 1908), present address unknown.

LATHROP, Jay Cowden (Affiliate, 1908), Supt. of Constr., Peoples Ry. Co., Dayton, O.

LUMMIS, Charles W. (1908), Mem. Am. Soc. M. E.

MANGELSDORFF, Max F. (Affiliate, 1910), 115 Nassau St., New York, N. Y., and *for mail*, 212 Fifth St., Union Hill, N. J.

QUINN, Stephen (Affiliate, 1909), Ch. Engr., Iola Portland Cement Co., and *for mail* Clinker Club, 16 N. Buckeye St., Iola, Kan.

ROTH, Charles (Affiliate, 1909), Mech. Engr., L. A. Becker Co., Chicago, and *for mail*, 220 Marion St., Oak Park, Ill.

THOMPSON, Wm. K. (Affiliate, 1909), 2619 Orchard Ave., Los Angeles, Cal.

NEW MEMBERS

JAMES, Frederick Conway (Affiliate, 1910), Cons. Engr., Mangold Bros., Port Elizabeth, and *for mail*, Steytlerville, Cape Colony, South Africa.

STUDENT BRANCHES

CHANGES OF ADDRESS

BINNS, G. W. (Student, 1910), 2358 Ohio Ave., Cincinnati, O.
FRAMBACH, F. S. (Student, 1910), Hartley Hall, Columbia Univ., New York, N. Y.
HAINES, P. G. (Student, 1910), 3153 Willis Ave., Cincinnati, O.
HENWOOD, P. E. (Student, 1909), 4620 Calumet Ave., Chicago, Ill.
HOLLENBERGER, Theo. J. (Student, 1909), 4433 N. Paulina St , Chicago, Ill.
KARMAZIN, John (Student, 1910), 1001 S. 5th St., Champaign, Ill.
LURIE, A. N. (Student, 1909), 1871 S. Kedzie Ave., Chicago, Ill.
MARSH, Karl H. (Student, 1910), with Arthur G. McKee, Engr., Rockefeller Bldg., Cleveland, O.
MURDUCK, R. K. (Student, 1910), 705 W. Hill St., Champaign, Ill.
YOAKUM, F. E., Jr. (Student, 1909), 201 Dryden Rd., Ithaca, N. Y.

NEW MEMBERS

BROOKLYN POLYTECHNIC INSTITUTE

BREVOORT, C. (Student, 1910), 69 Lincoln Rd., Flatbush, N. Y.

COLUMBIA UNIVERSITY

DAVIS, F. R. (Student, 1910), 220 W. 107th St., New York, N. Y.
DEMOREST, W. J. (Student, 1910), 173 W. 93d St., New York, N. Y.
SWALLOW, Howard (Student, 1910), 605 W. 184th St., New York, N. Y.
VAUGHAN, L. L. (Student, 1910), Hartley Hall, Columbia Univ., New York, N. Y.
VON MAHLENBORG, C. A. (Student, 1910), 127 W. 111th St., New York, N. Y.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

ESTES, G. H. (Student, 1910), 31 Newbury St., Boston, Mass.
MACKENZIE, Morril (Student, 1910), 68 Barnum St., Taunton, Mass.
ROBB, Charles A. (Student, 1910), 12 Oxford St., Cambridge, Mass.
RUSSELL, Foster (Student, 1910), 80 St. Botolph St., Boston, Mass.
STONE, R. T. (Student, 1910), Tech. Chambers, Boston, Mass.

UNIVERSITY OF CINCINNATI

COLBURN, B. V. (Student, 1910), 2343 Stratford Ave., Cincinnati, O.
HUMPHREYS, H. B. (Student, 1910), 236 Ofallon Ave., Bellevue, Ky.
LYTLE, C. W. (Student, 1910), 2345 Stratford Ave., Cincinnati, O.

UNIVERSITY OF ILLINOIS

CARLSON, C. A. (Student, 1910), 1015 W. Illinois St., Urbana, Ill.
JACOBSEN, C. H. (Student, 1910), 906 W. Illinois St., Urbana, Ill.
LINDSTROM, A. W. (Student, 1910), 307 E. Daniel St., Champaign, Ill.
PAUL, Harry (Student, 1910), 209 E. Green St., Champaign, Ill.

COMING MEETINGS

May–June

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AIR BRAKE ASSOCIATION

May 10–13, Dennisson Hotel, Indianapolis, Ind. Subjects for discussion, and chairmen: Air Brake Instruction, Examination and Rating, Thos. Clegg; Air Pump Piping, Fittings and Connections, George W. Kiehm; Best Arrangement of Air Pump and Main Reservoir Capacity for 100-car Train Service, P. J. Langan; Brake Cylinders and Connections to Cylinder Leakage, W. P. Garabrant; Inspection and Cleaning of Triple Valves and Brake Cylinders, C. P. McGinnis; Developments in Air Brakes, W. V. Turner; New York Brake Equipment, T. F. Lyons; Westinghouse Equipment, S. G. Down; Recommended Practice, S. G. Down, Secy., F. M. Nellis, 53 State St., Boston, Mass.

AMERICAN ASSOCIATION ELECTRIC MOTOR MANUFACTURERS

May 18, Newport News, Va. Secy., Frank H. Couch, Hampton.

AMERICAN CIVIC ALLIANCE

May 18, 29 W. 39th St., New York. Secy., Henry Frank.

AMERICAN EXPOSITION IN BERLIN

June 1-Aug. 31. American Manager, Max Vieweger, 50 Church St., New York.

AMERICAN FOUNDRYMEN'S ASSOCIATION
MANUFACTURERS SUPPLY ASSOCIATION

June 6–10, Detroit, Mich. Secy. of general committee, A. Preston Henry, Standard Pattern Works.

AMERICAN ELECTROCHEMICAL SOCIETY

May 4–7, Spring Meeting, Fort Pitt Hotel, Pittsburg, Pa. Among the papers will be: Furnace Conductors for Heavy Alternating Currents, K. C. Randall; A New Electric Steel Furnace, A. L. Queneau; Gases in Steel, P. L. T. Heroult; Cheap Power in the Pittsburg District, F. Crabtree; Induction Furnace Progress, T. Rowlands; Ductile Tungsten and Molybdenum, C. G. Fink; A New Process for the Treating of Cobalt-nickel Ores, C. C. Cito; A New Radiation Pyrometer, C. E. Foster, Mem. Am. Soc. M.E.; The Separation of Oil from Condenser Water by Electrolysis, H. M. Goodwin; A New Method for the Electrolytic Winning and Refining of Metals, E. M. Chance; Refining of Tin Dross in an Electric Furnace, R. S. Wile; The Effect of Moisture and of Solutions upon the Electrical

Conductivity of Soils, R. E. O. Davis; The Present Status of the Electro-Chemical Industries, J. W. Richards; Pittsburg as an Electrochemical Center, John A. Brashear, Mem.Am.Soc.M.E.; The Conservation and Utilization of Natural Sources of Power, J. H. Finney.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

June 22-24, summer meeting, Niagara Falls, N. Y. Secy., J. C. Olsen, Polytechnic Inst., Brooklyn.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

May 5-7, Auditorium of Home Telephone Co., 333 Grant Ave., San Francisco, Cal. Papers: Emergency Generating Stations for Service in Connection with Hydroelectric Transmission Plants under Pacific Coast Conditions, A. M. Hunt, Mem. Am. Soc. M. E.; Hydroelectric Power as Applied to Irrigation, J. C. Hays; The Developed High-Tension Net-Work of a General Power System, Paul M. Downing; Parallel Operation of Three-Phase Generators with their Neutrals Interconnected, G. I. Rhodes; Observations of Harmonics in Current and Potential Wave Shapes of Transformers, John J. Frank; Transmission Line Crossings of Railroad Right-of-way, A. H. Babcock. May 17, Annual Meeting; 33 W. 39th St., New York. May 27, special railway meeting, 33 W. 39th St. Papers: Application of Porcelain to Strain Insulators, W. H. Kempton; Electric Railway Overhead Construction, W. N. Smith. June 27-28, Annual Convention, Waumbeck Hotel, Jefferson, N. H. Secy., R. W. Pope, 33 W. 39th St.

AMERICAN PORTLAND CEMENT MANUFACTURERS

June, Kansas City, Kan. Secy., P. H. Wilson, Land Title Bldg., Philadelphia, Pa.

AMERICAN RAILWAY ACCOUNTING OFFICERS

June 29, Colorado Springs, Colo. Secy., C. G. Phillips, 143 Dearborn St., Chicago.

AMERICAN RAILWAY ASSOCIATION

May 18, 29 W. 39th St. New York, 11 a.m. Secy, W. F. Allen, 24 Park Pl.

AMERICAN RAILWAY INDUSTRIAL ASSOCIATION

May 10, Memphis, Tenn. Secy., Guy L. Stewart, S. W. Ry., St. Louis, Mo.

AMERICAN RAILWAY MASTER MECHANICS ASSOCIATION

June 20-22, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

AMERICAN SOCIETY OF CIVIL ENGINEERS

May 4, 18, 220 W. 57th St., New York. Papers, May 4: Water Supply of El Paso & Southwestern Railway from Carrizozo to Santa Rosa, New Mexico, J. L. Campbell; New York Tunnel Extension of Pennsylvania Railroad: Site of the Terminal Station, G. C. Clarke. Secy., C. W. Hunt.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

May 14, St. Louis, with coöperation of Engineers Club of St. Louis. May 31-June 3, Spring Meeting, Atlantic City, N. J. July 26–29, meeting with Institution of Mechanical Engineers, in Birmingham and London, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SOCIETY FOR TESTING MATERIALS

June 28-July 2, annual meeting, Atlantic City, N. J. Secy., Edgar Marburg, University of Pennsylvania, Philadelphia.

ASSOCIATION OF CAR-LIGHTING ENGINEERS

June 7, 8, semi-annual convention, Buffalo, N. Y. Secy., Geo. B. Colegrave, care of Central Railway, Chicago.

BROOKLYN ENGINEERS CLUB

May 5, clubhouse, 117 Remsen St., 8 p.m. Paper: Steel Centering as Applied to the Catskill Aqueduct, V. R. Whitehall. May 14, visit to Pa. R. R. Sta., New York. Secy., Joseph Strachan.

CANADIAN GAS ASSOCIATION

June 9-11, annual convention, Alexander Rink, Hamilton, Ont. Secy., A. W. Moore, Woodstock, Ont.

CENTRAL RAILWAY CLUB

May 13, Buffalo, N. Y. Secy., H. D. Vought, 95 Liberty St., New York.

CLEVELAND ENGINEERING SOCIETY

June 14, annual meeting, 714 Caxton Bldg. Secy., J. C. Beardsley.

ENGINEERS' CLUB OF BALTIMORE

June 4, annual meeting. Secy., R. K. Compton, City Hall.

ENGINEERS SOCIETY OF MILWAUKEE

June 8, annual meeting, Builders Club. Secy., W. F. Martin, 456 Broadway.

ENGINEERS SOCIETY OF PENNSYLVANIA

June 7, annual meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher, P. O. Box 704.

ENGINEERS' CLUB OF ST. LOUIS

May 14, coöperating with Am. Soc. M. E. Secy., A. S. Landsdorf, 3817 Olive St.

FOUNDRY AND MANUFACTURERS' SUPPLY ASSOCIATION

June 6–10, Detroit, Mich. Secy., C. E. Hoyt, Lewis Institute, Chicago.

FREIGHT CLAIM ASSOCIATION

June 15, Los Angeles, Cal. Secy., W. P. Taylor, Richmond, Va.

INTERNATIONAL CONGRESS OF INVENTORS

June 13–18, Rochester, N. Y.

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

Last week in June, Düsseldorf, Prussia. Secy., Dr. E. Schrödter, Jacobstrasse 315.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

May 24–26, New Clifton Hotel, Niagara Falls, Ont. Subjects for discussion and chairmen: Standardizing of Blue Prints, W. H. Laughridge; Application and Care of Flues, D. A. Lucas; Flexible Staybolts, C. J. Murray; Steel vs. Iron Tubes, M. O'Connor; Flue Holes in Back Flue Sheet, J. A. Dearnberger; Standardizing of Shop Tools, J. T. Goodwin; Standardizing of Pipe Flanges and Templates for Drilling, Jas. Crombie; Radical Departures in Boilers and Fire Boxes, B. F. Sarver; Fire Box Holes, W. H. Laughridge; Best Method of Staying Front Portion of Crown Sheet on Radical Top Boilers to prevent Cracking of Flue Sheet in Top Flange, H. J. Raps; Oxy-Acetylene Welding, M. O'Connor. Secy., H. D. Vought, 95 Liberty St., New York.

INTERNATIONAL RAILWAY FUEL ASSOCIATION

May 23–26, Chicago. Secy., D. B. Sebastian, 327 LaSalle St. Sta.

INTERNATIONAL RAILWAY GENERAL FOREMEN'S ASSOCIATION
May 3–7, Cincinnati, O. Secy., E. C. Cooke, Royal Ins. Bldg., Chicago.
IOWA DISTRICT GAS ASSOCIATION
June 15–17, annual meeting, Sioux City. Secy., G. I. Vincent, Des Moines.
LOUISIANA ENGINEERING SOCIETY
May 9, June 13, 321 Hibernia Bldg., New Orleans. Papers: The Manufacture of Sugar, W. H. P. Creighton; Coal in Relation to Boiler Economy, Wm. von Phul. Secy., L. C. Datz.
MASTER CAR BUILDERS ASSOCIATION
June 15–17, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.
MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK
May 27, inspection trip to Albany and Schenectady. Secy., C. D. Pollock, 29 W. 39th St., New York.
NATIONAL ASSOCIATION OF MANUFACTURERS
May 16–18, New York. Secy., George S. Boudinot, 170 Broadway.
NATIONAL DISTRICT HEATING ASSOCIATION
June 1–3, annual meeting, Toledo, O. Secy., D. C. Gaskill, Greenville, O.
NATIONAL ELECTRIC LIGHT ASSOCIATION
May 23–28, St. Louis, Mo., Secy., Frank H. Tate, Dayton, O.
NATIONAL ELECTRIC TRADES ASSOCIATION
June, San Francisco, Cal. Secy., F. B. Vose, 1343 Marquette Bldg., Chicago
NATIONAL FIRE PROTECTION ASSOCIATION
May 17–19, annual meeting, Hotel LaSalle, Chicago. Address by I. K. Pond, Pres. Am. Inst. Architects, etc. Secy., F. H. Wentworth, 87 Milk St., Boston.
NATIONAL GAS AND GASOLENE ENGINE TRADES ASSOCIATION
June 13–16, Semi-annual convention, Hotel Sinton, Cincinnati, O. Secy., Albert Stritmatter.
NATIONAL MACHINE TOOL-BUILDERS ASSOCIATION
May 24, 25, Spring Convention, Hotel Seneca, Rochester, N. Y. Secy., C. E. Hildreth, Worcester, Mass.
NATURAL GAS ASSOCIATION OF AMERICA
May 17–19, Oklahoma City, Okla. Secy., M. W. Walsh, 110 N. Broadway.
NEW ENGLAND WATERWORKS ASSOCIATION
June, Providence, R.I. September 14–16, annual convention, Rochester, N. Y. Secy., Willard Kent, Narragansett Pier, R. I.
NEW YORK RAILROAD CLUB
May 20, 29 W. 39th St., 8.15 p.m. Secy., H. D. Vought, 95 Liberty St.
OHIO SOCIETY OF ENGINEERS
May 19, 20, Cincinnati. Secy., F. E. Sanborn, State University, Columbus.
PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
May 24, West Hall, R. I. School of Design, 8 p. m. Paper: Modern Machine Tools. Second week in May, Spring visitation, The Norton Co., Worcester, Mass., June 28, annual meeting. Secy., T. M. Phetteplace, Mem. Am.Soc.M.E., 48 Snow St.
RAILWAY SIGNAL ASSOCIATION
June 14, 29 W. 39th St., New York, 9.30 a.m. Secy., C. C. Rosenberg, Bethlehem, Pa.

RENSSELAER SOCIETY OF ENGINEERS
June, annual meeting, Rensselaer Polytechnic Inst., 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

ST. LOUIS RAILWAY CLUB
May 13. Secy., B. W. Frauenthal, Union Station.

SOCIETY FOR PROMOTION OF ENGINEERING EDUCATION
June 23–25, Madison, Wis. Papers on Technical Education Abroad, Inspection Trips for Technical Students. Efficiency in Technical Education. Secy., H. H. Norris, Cornell University, Ithaca. N. Y.

STEVENS ENGINEERING SOCIETY
May 3, 10, Hoboken, N. J. Papers by C. F. Kroeh and J. A. Brashear, Hon. Mem.Am.Soc.M.E.

TELEPHONE SOCIETY OF NEW YORK
June 21, annual meeting, 29 W. 39th St. Secy., T. H. Woolhouse.

TRANSPORTATION AND CAR ACCOUNTING OFFICERS
June 28. Secy., G. P. Conard, 24 Park Pl., New York.

WESTERN CANADA RAILWAY CLUB
May 9, annual meeting, Royal Arms Hotel, Winnipeg, Manitoba. Secy., W. H. Rosevear, P. O. Box 1707.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
May			p. m.
4	Wireless Institute	S. L. Williams	7.30
5	Blue Room Engineering Society	W. D. Sprague	8.00
6	Western Union Electrical Society	H. C. Northen	7.00
7	Amer. Soc. Hungarian Engrs. and Archts	Z. deNemeth	8.30
12	Illuminating Engineering Society	P. S. Millar	8.00
13	Western Union Electrical Society	H. C. Northen	7.00
17	American Institute of Electrical Engineers	R. W. Pope	8.00
17	New York Telephone Society	T. H. Lawrence	8.00
			a. m.
18	American Railway Association	W. F. Allen	11.00
			p. m.
18	American Civic Alliance	Henry Frank	2.30 8.00
20	New York Railroad Club	H. D. Vought	8.15
20	Western Union Electrical Society	H. C. Northen	7.00
25	Municipal Engineers of the City of New York	C. D. Pollock	8.15
27	American Institute of Electrical Engineers	R. W. Pope	8.00
27	Western Union Electrical Society	H. C. Northen	7.00
June			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
3	Western Union Electrical Society	H. C. Northen	7.00
4	Amer. Soc. Hun. Engrs. and Archts	Z. deNemeth	8.30
9	Illuminating Engineering Society	P. S. Millar	8.15
10	Western Union Electrical Society	H. C. Northen	7.00

Date	Society	Secretary	Time
June			a.m.
14	Railway Signal Association	C. C. Rosenberg	9.30
			p. m.
17	Western Union Electrical Society	H. C. Northen	7.00
21	New York Telephone Society	T. H. Lawrence	8.00
24	Western Union Electrical Society	H. C. Northen	7.00

OFFICERS AND COUNCIL

PRESIDENT

George Westinghouse .... Pittsburg, Pa.

VICE-PRESIDENTS

Geo. M. Bond .... Hartford, Conn.
R. C. Carpenter .... Ithaca, N. Y.
F. M. Whyte .... New York

Terms expire at Annual Meeting of 1910

Charles Whiting Baker .... New York
W. F. M. Goss .... Urbana, Ill.
E. D. Meier .... New York

Terms expire at Annual Meeting of 1911

PAST PRESIDENTS

Members of the Council for 1910

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Frederick W. Taylor .... Philadelphia, Pa.
F. R. Hutton .... New York
M. L. Holman .... St. Louis, Mo.
Jesse M. Smith .... New York

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Wm. L. Abbott .... Chicago, Ill.
Alex. C. Humphreys .... New York
Henry G. Stott .... New York

Terms expire at Annual Meeting of 1910

H. L. Gantt .... Pawtucket, R. I.
I. E. Moultrop .... Boston, Mass.
W. J. Sando .... Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

J. Sellers Bancroft .... Philadelphia, Pa.
James Hartness .... Springfield, Vt.
H. G. Reist .... Schenectady, N. Y.

Terms expire at Annual Meeting of 1912

TREASURER

William H. Wiley .... New York

CHAIRMAN OF THE FINANCE COMMITTEE

Arthur M. Waitt .... New York

HONORARY SECRETARY

F. R. Hutton .... New York

SECRETARY

Calvin W. Rice .... 29 West 39th Street, New York

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Chas. Whiting Baker, *Vice-Chairman*
F. R. Hutton
H. L. Gantt
F. M. Whyte

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FINANCE

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Robert M. Dixon (3), *Vice-Chairman*
Edward F. Schnuck (1)
Geo. J. Roberts (2)
Waldo H. Marshall (4)

HOUSE

William Carter Dickerman (1) *Chairman*
Francis Blossom (3)
Bernard V. Swenson (2)
Edward Van Winkle (4)
H. R. Cobleigh (5)

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John W. Lieb, Jr. (3), *Chairman*
Leonard Waldo (2)
Ambrose Swasey (1)
Chas. L. Clarke (4)
Alfred Noble (5)

MEETINGS

Willis E. Hall (5), *Chairman*
L. R. Pomeroy (2)
Wm. H. Bryan (1)
Chas. E. Lucke (3)
H. de B. Parsons (4)

MEMBERSHIP

Charles R. Richards (1) *Chairman*
George J. Foran (3)
Francis H. Stillman (2)
Hosea Webster (4)
Theo. Stebbins (5)

PUBLICATION

D. S. Jacobus (1) *Chairman*
H. W. Spangler (3)
H. F. J. Porter (2)
Geo. I. Rockwood (4)
Geo. M. Basford (5)

RESEARCH

W. F. M. Goss (4), *Chairman*
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R. C. Carpenter (1)
Ralph D. Mershon (3)
Jas. Christie (5)

Note—Numbers in parentheses indicate number of years the member is yet to serve.

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
A. P. Trautwein — Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet — H. H Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman* — F. R. Hutton
G. M. Basford — D. S. Jacobus
Jesse M. Smith

On Conservation of Natural Resources

Geo. F. Swain, *Chairman* — L. D. Burlingame
Charles Whiting Baker — M. L. Holman
Calvin W. Rice

On International Standard for Pipe Threads

E. M. Herr, *Chairman* — Geo. M. Bond
William J. Baldwin — Stanley G. Flagg, Jr.

On Standards for Involute Gears

Wilfred Lewis, *Chairman* — E. R. Fellows
Hugo Bilgram — C. R. Gabriel
Gaetano Lanza

On Power Tests

D. S. Jacobus, *Chairman* — L. P. Breckenridge — Edward F. Miller
Edward T. Adams — William Kent — Arthur West
George H. Barrus — Charles E. Lucke — Albert C. Wood

On Student Branches

F. R. Hutton, Honorary Secretary

On Meetings of the Society in Boston

Ira N. Hollis, *Chairman* — I. E. Moultrop, *Secretary*
Edward F. Miller — J. H. Libbey
Charles T. Main

On Meetings of the Society in St. Louis

Wm. H. Bryan, *Chairman* — Ernest L. Ohle, *Secretary*
M. L. Holman

SOCIETY REPRESENTATIVES

1910

On John Fritz Medal

Ambrose Swasey (1) | Chas. Wallace Hunt (3)
F. R. Hutton (2) | Henry R. Towne (4)

On Board of Trustees United Engineering Societies Building

F. R. Hutton (1) | Fred J. Miller (2)
Jesse M. Smith (3)

On Library Conference Committee

J. W. Lieb, Jr., Chairman of the Library Committee, Am. Soc. M. E.

On National Fire Protection Association

John R. Freeman | Ira H. Woolson

On Joint Committee on Engineering Education

Alex. C. Humphreys | F. W. Taylor

On Government Advisory Board on Fuels and Structural Materials

Geo. H. Barrus | P. W. Gates
W. F. M. Goss

On Advisory Board National Conservation Commission

Geo. F. Swain | John R. Freeman
Chas. T. Main

On Council of American Association for the Advancement of Science

Alex. C. Humphreys | Fred J. Miller

Note—Numbers in parentheses indicate number of years the member is yet to serve.

OFFICERS OF THE GAS POWER SECTION

1909-1910

CHAIRMAN
J. R. Bibbins

SECRETARY
Geo. A. Orrok

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F. R. Hutton (3)
G. I. Rockwood (2)
H. H. Suplee (4)
F. R. Low (5)

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A. F. Stillman
H. V. O. Coes
G. M. S. Tait
A. E. Johnson
George W. Whyte
F. S. King
S. S. Wyer

GAS POWER MEETINGS COMMITTEE

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C. W. Obert
E. D. Dreyfus
W. H. Blauvelt
C. T. Wilkinson

GAS POWER LITERATURE COMMITTEE

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L. S. Marks
G. D. Conlee
T. M. Phetteplace
R. S. de Mitkiewicz
G. J. Rathbun
L. V. Goebbels
R. B. Bloemeke
L. N. Ludy
A. L. Rice
A. J. Wood

GAS POWER INSTALLATIONS COMMITTEE

L. B. Lent, *Chairman*
A. Bement
C. B. Rearick

GAS POWER PLANT OPERATIONS COMMITTEE

I. E. Moultrop, *Chairman*
C. N. Duffy
J. D. Andrew
H. J. K. Freyn
C. J. Davidson
W. S. Twining
C. W. Whiting

GAS POWER STANDARDIZATION COMMITTEE

C. E. Lucke, *Chairman*
E. T. Adams
Arthur West
James D. Andrew
J. R. Bibbins
H. F. Smith
Louis C. Doelling

Note—Numbers in parentheses indicate number of years the member is yet to serve.

OFFICERS OF STUDENT BRANCHES

INSTITUTION	BRANCH AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	....	F. E. Yoakum, Jr.
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	G. F. Gebhardt	F. E. Wernick	W. E. Thomas
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	A. F. Meston	J. B. Bubb
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. W. Templin	H. A. Houston
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	C. E. Johnson	C. A. Swiggett
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	B. L. Keown	C. S. Huntington
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	Chas. E. Lucke	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Morril Mackenzie	Foster Russell
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	W. H. Montgomery	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	John S. Langwill	Karl L. Kratz
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. E. Dudley	F. T. Kennedy
Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	M. E. Strieter	A. D. Stancliff
	1910			
Univ. of Maine, Orono, Me.	February 8	....	H. N. Danforth	A. H. Blaisdell
Univ. of Arkansas, Fayetteville, Ark.	April 12	....	....	....

See Trans. V. 31 - p. 3[illegible].

OPERATING EXPERIENCES WITH A BLAST FURNACE GAS POWER PLANT

By Heinrich J. Freyn

Member of the Society

The use of blast furnace gas engines in this country was first undertaken in 1902 by the Lackawanna Steel Company at Buffalo, followed four years later by the United States Steel Corporation in several of its plants. The import of the problem of utilizing the surplus gas may be realized by considering that eleven million tons of pig iron were produced in 1909 by the United States Steel Corporation, and that for each ton of iron made per day, 25 b.h.p. is available for purposes outside of the power requirements of the blast furnaces, provided this power is itself produced in gas engines. If, therefore, all the blast furnaces of the corporation were blown by gas blowing-engines and all other furnace requirements furnished by gas-electric engines, 750,000 b.h.p. would be available for other purposes.

2 In 1907 there were installed at one of the largest steel plants in this country, four Allis-Chalmers double-acting, four-cycle, twin tandem gas engines, gas cylinders, 42-in. diameter, 54-in. stroke, operating 2000-kw., 25 cycle, 3-phase, 2200-volt, alternating-current generators at 83.3 r.p.m. This addition to the existing steam-electric equipment was completed in 1908 and the electric power produced by these gas engines is used for electric-driven rolling mills and general light and power purposes. The gas-driven generators operate parallel with the adjacent steam units and with other gas-driven generators located 20 miles away.

3 The gas engine plant under discussion has been in regular service for one and one-half years, during which period the experiences and results described in the following pages were obtained.[1] These

[1]These experiences and records were compiled with the able assistance of Mr. Chas. C. Sampson, Mem.Am.Soc.M.E.

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

were not gathered from the indications of one single experiment, or of a series of carefully prepared and conducted tests, but represent the average results of daily observations extending over one year's time. Since the degree of correctness depends on the accuracy of observation and care in recording the results of persons having various degrees of practical and technical training, inaccuracies entailing puzzling inconsistencies may have crept in and the data presented may not in every instance withstand the test of searching criticism; nevertheless it is believed that such information, derived from actual operation, will prove of more interest to the engineering profession than unassailable data obtained under test conditions.

4 The gas supply for the operation of all the gas engines of the plant is taken from six blast furnaces, all of which in 1909 were blown by steam blowing-engines, while the electric power for the plant was derived from both steam and gas-driven generators. This plant is therefore a so-called "mixed" plant, so far as the generation of power is concerned.

5 The quantity of blast furnace gas available for the operation of gas engines was therefore considerably less than it will be when four of these furnaces are blown by gas blowing-engines. Because of the general business depression at the beginning of 1909, only three, and in the months of March and April only two, furnaces were in blast. Normal conditions were resumed about May or June, while all six furnaces were in blast during the months of September and October only.

CONDITIONS OF INSTALLATION

6 The gas power station in question was conceived in 1905, and all preliminary calculations relative to the amount of gas available for operating gas engines were naturally based on the conditions existing at that time, making the proper deductions for reduced gas production due to the furnaces being out of blast for relining. Careful investigations showed that in 1906, 10 per cent of the total gas produced by six blast furnaces, equivalent to 10,200 kw., was available for use in gas-electric engines. The installation of 8000 kw. in gas engines seemed therefore fully warranted, particularly as it was expected that two gas blowing-engines simultaneously ordered would be in operation after November 1907, in which case the gas surplus, even with only five furnaces in blast (one furnace down for relining) would have been more than ample to operate four 2000 kw. of gas-electric units. It could not be foreseen that business conditions would

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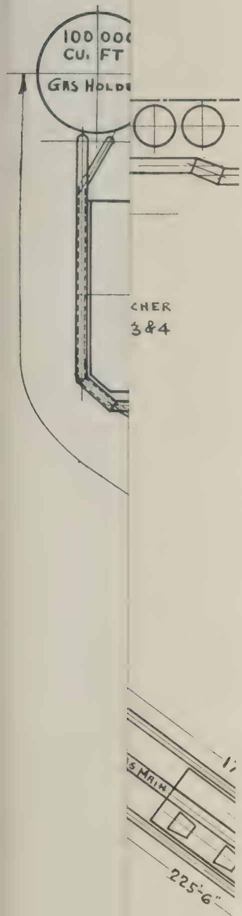


NT 1910

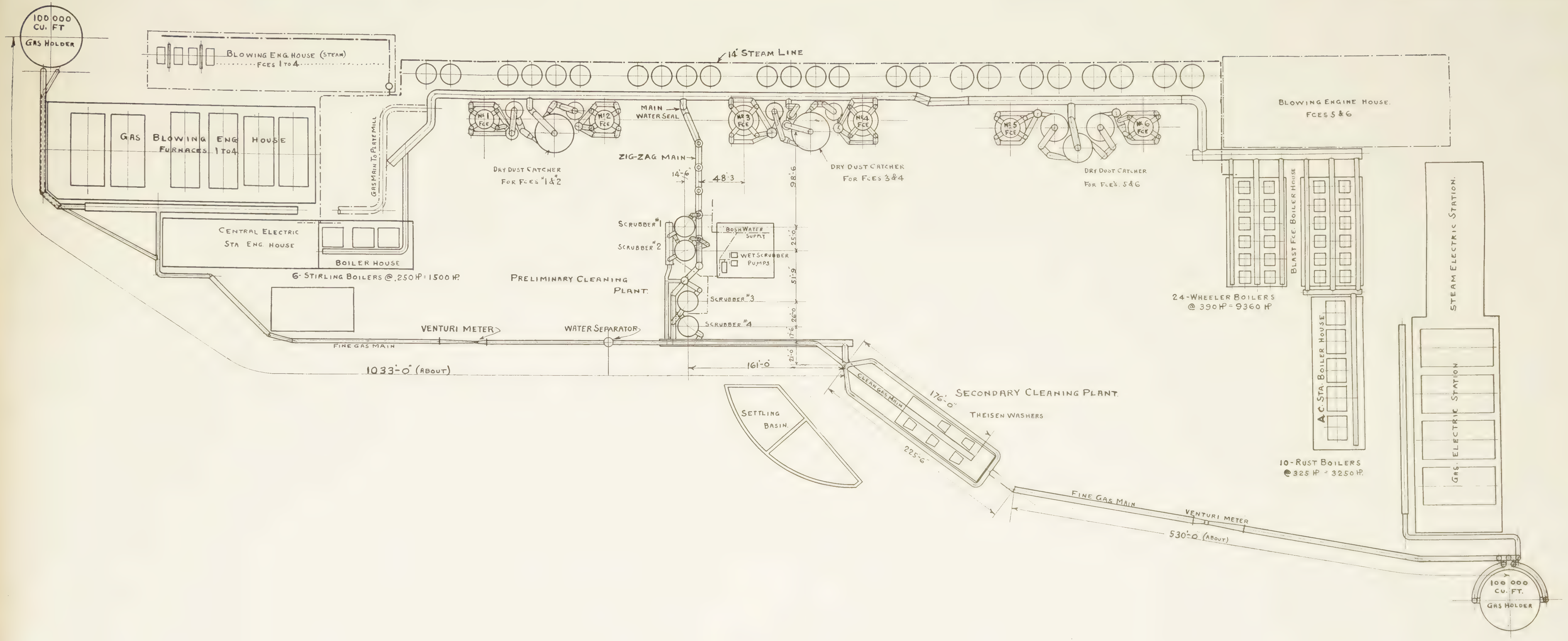


FIG. 15 ARRANGEMENT OF GAS POWER PLANT 1910

change so radically in 1908, nor that the two gas blowing-engines would be so delayed, that for three years the saving of gas, which would have materially improved conditions for the electric units, could not be realized.

7 While the logical way to begin would have been with the installation of a number of gas blowing-engines, instead of first taking gas-electric engines in operation, increasing instead of reducing the available quantity of blast-furnace gas, such a procedure was impossible because of the immediate demand for increased electric power created by the installation of new electrically operated mills, as well as on account of local conditions of steam supply for the furnaces, which at that time prohibited the removal of a large boiler house, now occupied by the new gas blowing-engines. From the circumstances, however, that gas-electric engines were installed before any gas blowing-engine equipment existed, and that this power plant, as it so happened, had to be operated for almost two years under the most unfavorable and exacting conditions, a great deal of most valuable experience was gained, in that it was found that such a power plant could be maintained in operation—although with interruptions—with only two furnaces, and for a short period even with only one furnace in blast.

OUTPUT OF POWER PLANT

8 In Appendix No. 1, Table 1 shows the average kilowatt-hour produced by the gas power plant for each month of 1909, from which it appears that the average for the year was 5760 kw-hr., or 72 per cent of the total capacity of the plant, and this average for various months varied from 66.5% to 74% with two furnaces, 61.5% to 80.5% with three, 69% for four, 64% to 82.5% with five, 68.5% to 78% with six furnaces in blast. While during the first few months the number of furnaces in blast was very limited, the total output of the station was nevertheless not affected very materially. In fact, in the month of March, when only furnaces No. 1 and No. 2 were in blast, 74% of the total capacity of the plant was produced, a higher figure than the average output of the plant for the whole year.

SHUTDOWNS AND TIME LOST IN OPERATION

9 A record is being kept as correctly as practicable of all shutdowns and their causes. Table 2, in Appendix No. 1, gives the monthly averages, as well as the percentages of operating time and of time lost chargeable to the engines, and due to outside causes. The

power station is considered in this table as one unit, and the figures are averages applying to the four engines.

10 In Appendix No. 1, Table 2, is given the average monthly operating time of the station for the whole year, from which it appears that the average for the year was 77% of the total possible time, 14.2 % and 8.8% being the percentages of the time lost due to engine repairs and to outside causes beyond control. The respective figures for the first four months of the year show that the operating time from January to April was much lower than during the rest of the year, the lowest figure being 57% in January. However, the month of March with 71% shows again the noteworthy fact that with only two furnaces in blast the operating time was higher than the average for the first half of the year. This was made possible only by shutting off the boiler houses almost entirely. With the boiler houses off, blast furnace No. 1 made more gas than the gas engines, the stoves and one small boiler house could use, so that one bleeder at the furnace had to be kept open. During casting periods the engines were operated on the gas tank. In this manner operation was kept up for one whole week. The time lost chargeable to the engines is considerable for the first four months, due to the fact that owing to the uncertainty of sufficient gas supply under the existing conditions of furnace operation certain repairs and alterations were made on the engines, which otherwise would have been distributed over a longer period of time. It is to be noted that any time lost is rigorously charged against the engines if the latter for any reason are not ready to resume operation at any moment.

11 The time lost due to outside causes was particularly heavy in the first four months of the year, varying from $11\frac{1}{2}$% in April to $19\frac{1}{2}$ % in February. In the records the lost time chargeable to outside causes is subdivided into losses due to operation of the plant, such as line troubles, or output not required, etc.; and losses due to lack of gas. This particular information is given in Table 3 of Appendix 1, wherein the plant is again considered as one unit. Shortage of gas was responsible to the greatest extent for lost time from outside causes in the first four months of the year. In January this figure was as high as 94.5 %, and while the average for the first half of 1909 exceeds 60%, the corresponding figure for the second half of 1909 is only 3%.

CONSIDERATIONS OF SAFETY WHEN THERE IS SHORTAGE OF GAS

12 Although the difficulties which were experienced in this period by the frequent inability of the blast furnaces to supply sufficient gas

to maintain operation of the whole plant, without heavily firing coal under the boilers, had no serious effect on the gas power plant, one important question was in the minds of all during this period, namely the safety of the installation. Antedating only a few months this period of gas shortage, a serious accident had happened at another plant where through lack of gas while only one furnace was in blast, the preliminary cleaning plant, the gas holder and parts of the pipe line conveying gas to the engines, exploded with disastrous effect. This accident caused a great deal of uneasiness and alarm in other gas engine plants where several furnaces were out of blast.

13 A gas power plant is endangered in two ways by lack of gas, either from collapsing of the gas holder bell or from explosion. In modern gas-cleaning installations, the so-called secondary washing plant, which refines the gas for use in engines, is usually equipped with some kind of rotary washers. Certain washers of this type, such as the Theisen, can give a vacuum of 3 in. of water, and a discharge pressure from 8 in. to 10 in. higher than the positive or negative pressure on the suction side. The washers deliver the gas to a gas holder under variable pressure dependent upon the raw gas pressure, while it is the principal object of the gas holder to maintain a constant gas pressure at the gas engines, irrespective of what the pressure at the blast furnaces or in the gas-cleaning plant happens to be. As long as the pressure of the gas, and therefore its quantity, is sufficient to allow the rotary washers to keep the gas-holder bell floating; in other words as long as balance exists between the demand for gas on the part of the engines and the supply from the furnaces, there is no danger to the installation.

14 If the gas supply falls below the demand, the volume of gas in the holder will cover the shortage within the limit of its capacity and until the bell, descending completely, rests on its landing beams. The rotary gas washers will then continue to operate, creating a depression in the gas conduits by which they are connected to the gas main at the furnaces. The latter is virtually a large gas receiver into which all blast furnaces discharge their gas, and which in turn supplies the hot-blast stoves, the boilers, and the gas-cleaning plant. The vacuum created by the rotary washers will naturally be communicated to this main gas flue, but cannot be maintained as the overhead flue is connected with the atmosphere through hot-blast stoves and boiler stacks. Air will therefore rush into this flue and into the gas-cleaning plant and be drawn into the rotary washers together with whatever gas is supplied, and discharged into the gas holder.

As long as these conditions exist, the gas-holder bell is not in danger from collapsing, but there is imminent danger from explosion to the whole plant. Rotary gas washers do not discriminate between gas and air, and continue to operate, filling all gas flues, gas holder and engine connections with a mixture of gas and air which under certain conditions is highly explosive. Should backfiring occur in the gas engines when receiving a mixture of gas and air instead of pure gas; that is, should the fresh incoming charge accidentally be ignited, consequences would be as prompt as disastrous —the air and gas mixture in the pipe system would explode, possibly wrecking the whole installation by a series of explosions.

15 This is precisely what did happen in the accident mentioned, and profiting by this experience steps were taken to prevent the occurrence of such an accident at the plant under discussion. Power house, gas washing plant and blast furnace office were connected by two independent telephone lines, and recording instruments, in addition to ordinary U-tubes, were installed in the washer building and at the blast furnace office, so that not only may the gas pressure be observed at any time, but it is automatically recorded for each period of 24 hours. Moreover an automatic alarm was installed at the blast furnace office, which rings a bell as soon as the gas pressure in the raw gas descends below a certain danger point, and whistle signals operated by solenoids from the blast furnace office were provided in the boiler house to inform the head fireman of the number of boilers to be "taken off" or put on gas. In addition an automatic bell was placed in this boiler house, calling the operators' attention to any drop below normal in the gas pressure.

16 Independently of the blast furnace department, the gas-cleaning plant operators were also carefully watching the gas pressure. The position of the gas-holder bell was made visible at any time by a system of incandescent lamps in the washerhouse, and strict orders regarding the use of the gas were issued by the blast furnace superintendent, instructing the men to favor the gas engines under any circumstances, as it was fully recognized that having taken care of the requirements of the hot-blast stoves, the remaining gas could not possibly be more efficiently utilized than in the gas engines. The practice was to shut off the gas immediately at a certain number of gas-fired boilers, as soon as the pressure in the overhead gas flue dropped below a predetermined point. Additional boilers were taken off if the gas pressure did not recover, so that sometimes as many as 24 boilers were being fired by coal exclusively. If this did not have

the desired result, stoves were taken off for short periods to increase the gas pressure above the danger point. At last, if all these steps did not improve the situation, one or more gas engines were shut down.

17 Fortunately in the majority of cases the blast furnace operators know in advance if the gas supply is likely to fail, and communication could easily be established to warn the departments concerned of the impending gas shortage. The diagram, Fig. 1, plotted from a

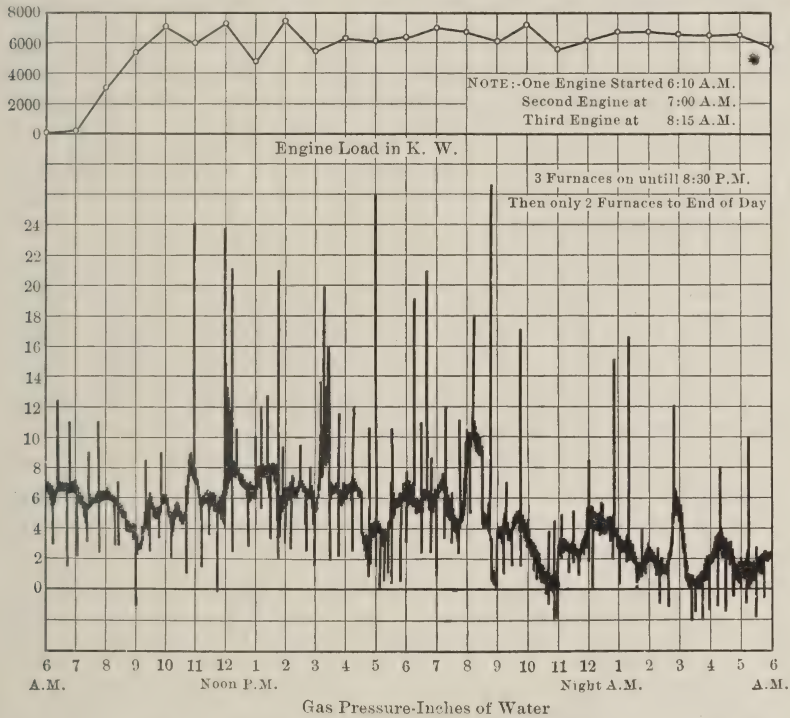


Fig. 1 Diagram Plotted From Bristol Chart Showing Gas Pressure where the Gas Enters the Gas-Cleaning Plant

Bristol chart representing a 24-hour record of the gas pressure at the point where the gas enters the gas-cleaning plant, is a good illustration of the conditions existing on many occasions.

18 The system of close observation and of coöperation among the departments concerned worked to perfection, but nevertheless conditions existed at times which with all due optimism had to be called dangerous. It was frequently necessary to keep several gas engines running, with the gas pressure dropping below the danger point

momentarily or even for periods of a few minutes. This was unavoidable if the operation of certain departments dependent upon a supply of electric power was to be maintained with any regularity. If the gas engines had been shut down every time a momentary drop in pressure occurred, it would often have meant an endless amount of shutting down and starting of engines, altogether too frequent for satisfactory operation of the mills, and physically impossible for the gas engine operators.

19 The question whether an automatic safety device should be installed at the power plant under discussion was thoroughly considered and such devices were investigated; but it was decided that the installation of costly safety appliances, which were certain to become inoperative with the normal number of furnaces in blast, was not warranted, as the conditions of gas shortage were exceptional and of temporary nature only. Besides, automatic safety devices, no matter how ingeniously designed, are never "foolproof," and have the reputation of operating without cause and of failing to act when needed. A further drawback is the tendency to over-confidence in the infallibility of a safety device. In this respect gas-cleaning plants should be classed with boiler plants, where the "human element" cannot be eliminated, and safety depends ultimately upon the rigid enforcement of certain established regulations. Responsible operators can use good judgment which automatic safety devices do not possess to decide whether shutdowns are necessary when low gas pressure occurs, possibly for a moment only. This was proved time and time again at the plant under discussion.

20 If such safety devices are considered necessary, however, the arrangement of automatic circuit breakers to shut off the power at the rotary washers, and simultaneously interrupt the ignition circuit of the gas engines, is decidedly better for safeguarding the plant than the installation of butterfly or check valves between rotary washers and gas holder, which shut off delivery under the control of the gas pressure. While in both cases the aspiration of air by the rotary washers is effectively prevented, the former device protects not only the gas cleaning plant but also the gas holder, while the latter may cause collapsing of the holder bell by isolating it from the gas supply.

QUANTITY AND QUALITY OF GAS SUPPLIED TO ENGINES

21 The amount of gas produced by each blast furnace is calculated and distributed in proper proportion among the different places of its consumption. Monthly gas-distribution sheets give a record of

the average daily tonnage of each furnace, the kind of blast, whether natural or dry, the kind of coke used and the coke consumption per ton of iron; further, the average gas analysis for each furnace based on daily determinations of continuous 24-hour samples, the heat value per cubic foot at 62 deg. fahr. and including the sensible heat of the gas at 500 deg. fahr., the temperature of the air at the blowing engines, the number of cubic feet of air blown per minute, and the average blast pressure. From these data the quantity of gas produced by each furnace per minute is calculated according to methods given in Appendix No. 2. The distribution of the gas from one blast furnace based on such calculations is given in the accompanying table (Table 1 herewith), reproduced from data given in Appendix No. 2.

TABLE 1 DISTRIBUTION OF GAS FROM BLAST FURNACE NO. 6

AUGUST, 1909

	Million B.t.u.	Per Cent
Total Gas Generated	324.1	100
Stoves and leakage	130.0	40.
Blowing engines	92.1	28.4
Used at furnace	9.0	2.8
Auxiliaries	4.6	1.4
Total used for blast furnace operation	235.7	72.6
B.t.u. surplus for furnace	88.4	27.4
B.h.p. equivalent of surplus	1470	

22 An excellent practical indicator of the gas quantity available for engine operation is the gas pressure at the cleaning plant. With more than three furnaces in blast the pressure is always sufficiently high to make operation of the gas power station perfectly safe. Fig. 2 shows the average monthly gas pressure at the main water seal

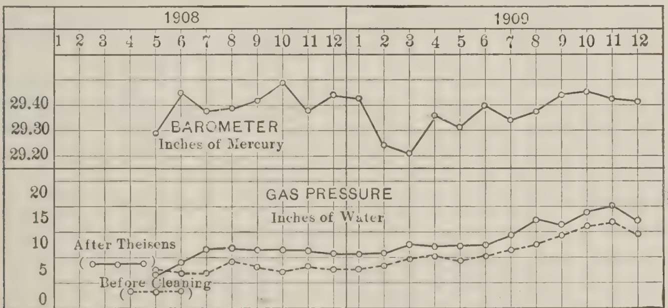


FIG. 2 GAS PRESSURE CURVES—BAROMETRIC PRESSURE (MONTHLY AVERAGES)

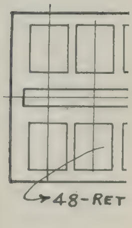
where the gas enters the cleaning plant and in the fine gas main after the secondary washers.

23 While the quantity of blast furnace gas was subjected to considerable variation due to the generally unfavorable conditions which existed in the early part of 1909, the quality of the gas was also found to vary materially, so far as its chemical composition and heat value were concerned. These were influenced not only by changes in the furnace burden and by the kind of product, whether basic iron, Bessemer iron, spiegeleisen or ferrosilicon, but by other causes to which variations frequently recorded from hour to hour, to a large extent could be traced. The blast furnaces discharge their gas into one common overhead gas main supplying stoves, boiler houses and gas engines. The intake for the gas-cleaning plant is located between furnaces No. 2 and No. 3, dividing the total length of the overhead flue in the proportion of one to two, approximately. In view of this central location of the intake nozzle it was expected that by mixing the gas from these furnaces a fairly uniform quality, representing the average of all six furnaces, would be obtained for engine operation. Due to the location of the boiler houses, however, this uniformity of mixture could not be realized.

24 Fig. 3 shows the existing conditions previous to May 1909, and before the boiler plant for four blast furnaces was abandoned in order to make room for the new gas blowing-engine house. The tall boiler stacks caused a flow of gas from furnaces No. 1 and No. 2 to the boiler house situated at the west end of the flue, while the gas from furnaces Nos. 4, 5 and 6 went to two large boiler houses located at the opposite extremity. The gas-cleaning plant received, therefore, almost exclusively gas from furnaces No. 2 and No. 3 or from No. 3 alone, while No. 2 was out of blast. This was proved beyond any doubt by frequently comparing the chemical analysis of the gas delivered at the power station with the analyses of the gas of the individual furnaces.

25 Thus for the month of June 1908 the average composition of the gas from blast furnaces Nos. 1, 3 and 4, which are in close proximity to the gas-cleaning plant intake, was as follows (blast furnace No. 2 being out of blast and blast furnace No. 4 on spiegel):

Blast furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.
1	13.5	25.0	3.4	1.85	93.3
3	12.4	26.7	3.2	2.15	98.1
4	5.1	32.1	3.3	6.30	115.2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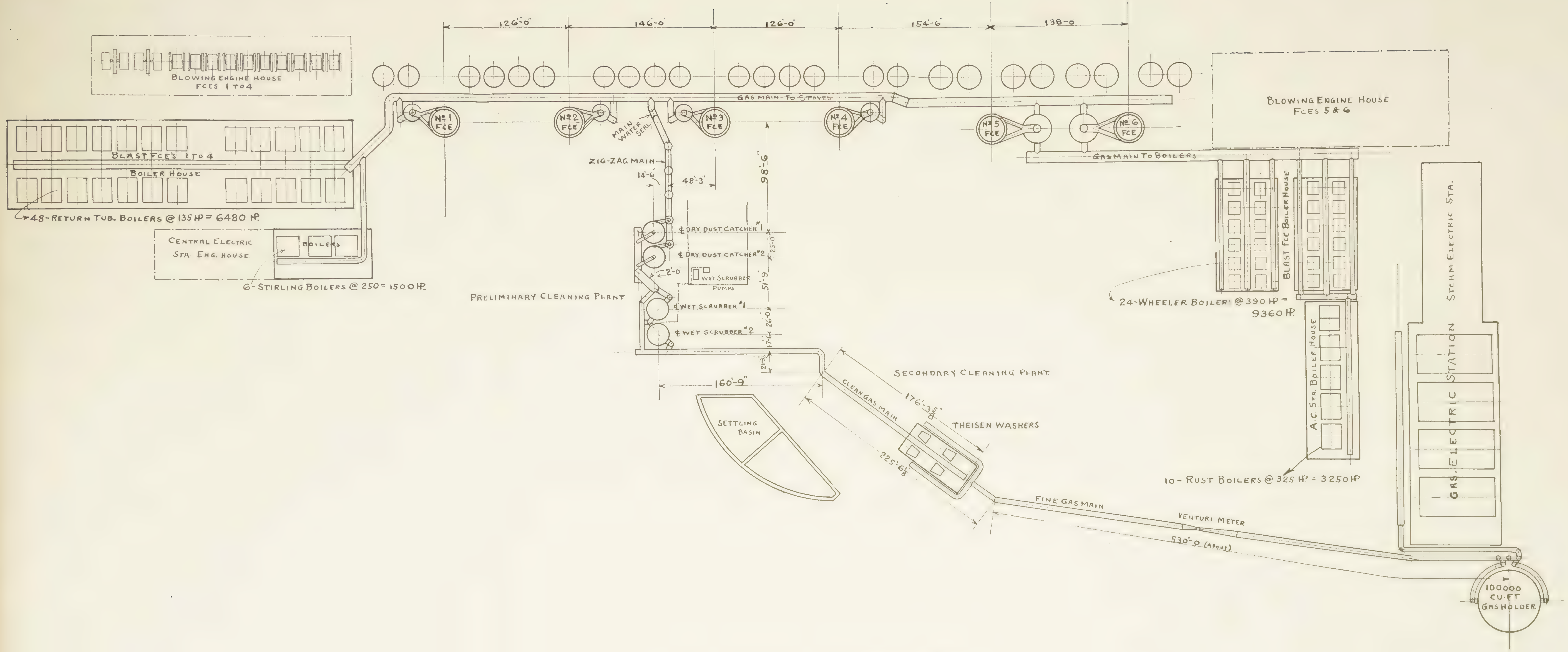


FIG. 3 ARRANGEMENT OF GAS POWER PLANT 1908

The average composition of the gas at the power station for the same period was:

CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.
12.6	26.64	3.2	2.33	96.73

The analyses of the gas of blast furnace No. 3 and the gas at the power house, coincide very closely, establishing proof that during this time this one furnace was furnishing the gas for the engines almost exclusively.

26 Since abandoning the boiler house for blast furnaces Nos. 1 to 4, which now receive steam through a 14-in. steam line from the boiler house at the east end, all gas from furnaces Nos. 1, 2 and 3 is being delivered to the gas-cleaning plant, with the exception of a small portion which goes to a small boiler house at the west end, while the gas of the remainder of the furnaces flows in the same direction to the boiler houses of furnaces Nos. 5 and 6 and the electric station. To illustrate the present condition of gas distribution, comparative data were compiled, given in Tables 3 and 4 in Appendix 2, which give the averages of gas analysis and heat values of the individual six furnaces, all of which were in blast in September 1909 and the average composition of mixtures of the gas from various furnaces calculated from the former. For the same month when these averages were taken the average composition and heat value of the gas delivered at the power house were:

CO_2	CO	H	CO_2	B.t.u.
10.03	29.80	3.77	2.98	108.70

Comparing this analysis with the mixture characteristics, given in Table 4, Appendix 2, the gas appears to be most nearly equivalent to the mixture from furnaces Nos. 1, 2 and 3 together.

27 Conditions were decidedly better in the second half of 1909, so far as uniformity of the gas supplied to the engines is concerned, but it is easily seen that changes in furnace operation must even under present conditions affect the quality of the engine gas. Whenever the gas supply from the furnaces on which the gas-cleaning plant is directly drawing happens to cease, in other words during checks or repairs, or if one or several of these furnaces are in trouble, disturbances are created in the regular flow and therefore in the quality of the gas, so that momentarily, or for longer periods, richer or leaner gas from other furnaces near the gas-cleaning plant intake is delivered to the gas engines. That such disturbances exist was very strikingly

proved in many instances. The gas engines, which had been operating smoothly, apparently receiving very uniform gas, would suddenly begin to backfire, or to have premature explosions and become very unsteady. These abnormal occurrences would be repeated at short intervals, although possibly lasting only a few minutes. The operating engineers soon discovered the cause of their troubles, and reported in their language that "a bad batch of gas" had caused the backfiring or the premature explosions and the "swinging" on the line. Such pronounced "waves" in the quality of the gas will often affect first the engine nearest to the gas holder, the trouble gradually extending to the engines down the line, and will stop first at the engine where the trouble started, gradually lessening on the rest of the engines, or else all engines will be affected simultaneously.

28 These interesting phenomena, and their bad effect on the parallel operation of the gas engines, prompted investigations which almost invariably located the causes for the sudden increase in hydrogen and methane. It was found that slipping of the furnaces was very frequently followed by backfires and premature explosions; and whether or not part of the raw wet stock in the furnaces reaches the incandescent zone, due to the upheaval of the material inside the furnace during slipping, thereby causing the formation of excessive amounts of hydrogen, remains an open question. Violent premature explosions and backfiring could in very many cases also be traced back to leaking tuyeres or hot blast valves, and these were so pronounced at times that the gas engines often fairly served as an indicator of such leaks.

29 The following gas analyses made on February 10, 1909, at the power house, give a good illustration of the suddenness of these changes:

GAS ANALYSES AT POWER STATION LABORATORY

Engine Gas

Time	CO_2	CO	H	CH_4	B.t.u. by analysis
11.00 a.m.	14.9	24.5	3.5	0.2	90.9
12.30 p.m.	13.8	24.7	4.3	0.3	94.6
3.30 p.m.	14.5	25.0	6.5	0.1	99.9
4.10 p.m.	14.2	24.8	4.5	0.2	94.7

The increase in hydrogen of almost 100 per cent between the first and third analyses is noteworthy. The effect on the engines was the occurrence of violent premature explosions around 3 o'clock of that

day. Backfiring happened simultaneously on three engines in operation on August 17, 1908, and was caused by fluctuations in the composition of gas. The daily chemical report for 24 hourse nding 6.00 a.m., August 18, contains the following record:

CHEMICAL ANALYSIS, AUGUST 18

Time	CO_2	CO	H	CH_4	B.t.u. by analysis
9.30 a.m.	10.9	27.6	3.6	0.2	101.2
1.30 p.m.	6.4	33.2	4.4	0.2	121.6
2.00 p.m.	8.4	30.2	4.4	0.2	111.9
4.00 p.m.	8.0	30.2	3.5	0.2	109.4

30 The simultaneous calorimeter determinations gave the following heat values:

CHANGE IN HEAT VALUE BY CALORIMETER

Time	B.t.u.	Time	B.t.u.
9.00 a.m.	101.3	1.00 p.m.	115.2
9.30 a.m.	102.7	1.30 p.m.	118.3
10.00 a.m.	103.9	2.00 p.m.	110.7
10.30 a.m.	105.0	2.30 p.m.	110.3
11.00 a.m.	105.9	3.00 p.m.	112.0
11.30 a.m.	106.4	3.30 p.m.	108.5
12.00 m.	108.6	4.00 p.m.	109.3
12.30 p.m.	109.5	4.30 p.m.	110.4

The heat value of the gas increased almost 20 B.t.u., or about 20 per cent in less than three hours, due to heavy coke blanks charged at blast furnace No. 1 which was in trouble. In this particular instance it was the sudden increase in carbon monoxid which caused the backfiring. It was not always possible, however, to prove by analysis or by calorimeter test that a sudden change in the heat value of the gas or a momentary increase in hydrogen had taken place when premature firing occurred; nevertheless, following the example of the Lackawanna Steel Company, the pressure of the cooling water for tuyeres and hot blast valves was reduced to a little below the normal blast pressure on all furnaces. Thus water-leaks into the furnaces were very effectively stopped and one principal cause for premature explosions at the engines was removed.

31 The kind of iron produced by the different furnaces at different times had a considerable effect upon the quality of the gas. Thus in September 1909 the average heat value of the gas at the power station was 108.7 B.t.u. per cu. ft., because during this month blast furnace

No. 1 was making ferrosilicon with a coke consumption of over 4600 lb. per ton of product. The average analysis of the gas of this furnace is given in Table 3, Appendix 2, and the composition of the engine gas for September is given in Par. 26 of the paper. The richest gas which the engines ever received occurred September 17, 1909, the average of the analyses for the day being as follows:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
4.7	34.9	3.2	0.16	7.92	123.3

The average corresponding B.t.u. values determined by calorimeter were 122.5 per cu. ft. The influence of this rich gas on the operation of the engines is shown in the daily record of engine operations for the same day. Three engines were running and were backfiring and hav-

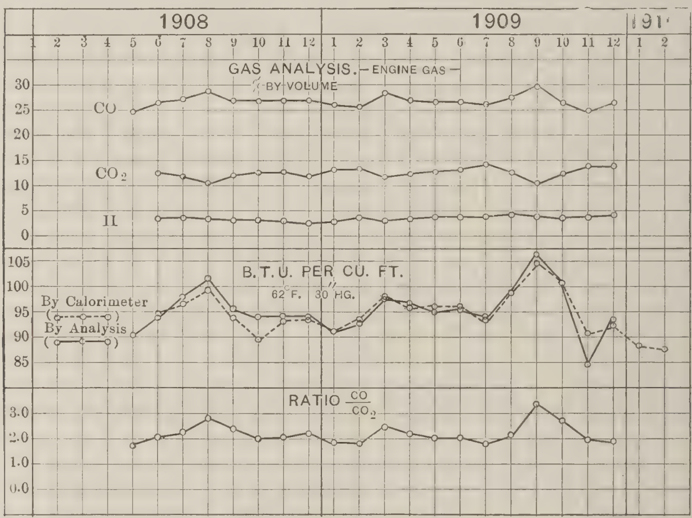


FIG. 4 GAS ANALYSIS AND HEAT VALUE OF BLAST FURNACE GAS (MONTHLY AVERAGES)

ing premature explosions all day long. On the following day the operating engineer reported that the gas made a quick change about 1.00 a.m. to very poor quality, causing all engines to misfire and to drop about one-half of their load, and the richness of mixture had to be changed on all engines to obtain proper ignition. About 2.00 a.m. the gas became suddenly very rich and the engines again backfired heavily, necessitating additional changes in the mixture. The leanest gas

on which these engines were operated during the year 1909 occurred in November, the average composition for the month being as follows:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u.
13.8	24.7	3.59	0.19	1.79	86.7

The lowest daily average heat value occurred on November 16, with gas of the following analysis:

CO	CO_2	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
15.8	22.1	3.3	0.3	1.39	83.1

The gas was so poor that day that it was impossible to keep it burning in the calorimeter. The lowest heat value ever recorded is 79.5 B.t.u. per cu. ft. on November 17, the gas analysis at 12.00 o'clock noon giving the following results:

CO_2	CO	H	CH_4	$\frac{CO}{CO_2}$	B.t.u. by analysis
17.1	21.6	3.1	0.1	1.26	79.5

The effect of such very poor gas on the engines is quite noticeable; the full output of the generators could not be maintained, although the proportion of air and gas was changed to meet the new conditions.

32 In Appendix 2, Table 5, is given the average composition of the blast furnace gas for each month, averages for the first and second halves, and the average for the whole year of 1909; further, the heat value of the gas determined by calorimeter. In Fig. 4 herewith these values are plotted for each month since June 1908 when the systematic records were begun. The discrepancies in the heat values as computed, and as determined by Junkers calorimeter, are explained by the fact that analyses are made about every three hours, while calorimeter readings are taken almost continuously during the day. The number of observations is therefore much greater for the latter than for the former. For all calculations the heat values determined by calorimeter are used exclusively. The methods of gas sampling and analysis used as described by Mr. L. A. Touzalin are as follows:

Daily samples of blast furnace gas are taken at each individual furnace, all samples being accumulative and representing a fair average of the gas production extending over a period of 24 hours. The sample is taken between down comerstand pipe and dust catcher (as shown in Fig. 5) and conducted by means of a $2\frac{1}{2}$-in. or 2-in. pipe, first to a miniature dust catcher and then to two washing bottles connected in series by means of a 2-in. pipe. From the second bottle

the gas passes to a sampling tank of 5 cu. ft. capacity, made of galvanized iron and of regular gas-holder construction. All water used in the washing bottles and in the sampling tank is first saturated with blast furnace gas. The small dust catcher consists of an 18-in. length of 6-in. iron pipe capped at each end and suspended in a vertical position. By removing the cap at the bottom end, the accumulated dust may be cleaned out as often as necessary. The valve placed at the lower end permits a small stream of gas to flow continuously through the sampling pipe into the small dust catcher and escape into the air. A great deal of dust escapes with this gas, and besides preventing continuous clogging, this valve allows any condensed water to drain off. The screw cock attached to the rubber tube between the washing bottles affords means of so adjusting the rate of flow that the tank is almost filled in 24 hours. At the end of each period a sample of the accumulated gas in the tank is withdrawn into a glass gas holder of 250 cu. cm. capacity, while the remaining gas in the tank is allowed to escape into the air. The bell of the tank thus drops down in place and the gas is again started for the next 24-hour sample. The 250 cu. cm. sample of the gas in the gas holder is taken to the laboratory for analysis. At the power station, where a special gas laboratory, fully equipped, is installed, gas samples are taken directly from the gas main between gas holder and engines. Gas analyses are made several times during the day and as often as necessary if any unusual occurrences at the engines indicate a change in gas quality.

33 Daily gas analyses and heat values as well as results of dust and moisture determination, together with additional chemical information, are recorded on daily report sheets (Fig. 6).

DESCRIPTION OF GAS-CLEANING PLANT

34 When the installation of blast furnace gas engines was decided on in 1906, very little information and experience on the important matter of gas cleaning was available in this country. Some experiments had previously been made at the plant under discussion on a small scale, with different designs of wet scrubbers and baffle washers, the deception generally prevailing at that time that gas could be cleaned sufficiently for engine purposes by so-called "static" methods, that is, by passing it through towers filled with baffle plates and a checker work of wood or iron, and sprayed with water in finely divided form. The results of these experiments were discouraging, as might have been expected, and the installation of Theisen gas washers for refining the gas was eventually decided upon.

35 The blast furnace gas for the gas engines is cleaned in two distinct stages. It is first subjected to a preliminary dry and wet scrubbing in the so-called primary or preliminary cleaning plant, and subsequently undergoes a secondary cleaning or refining by Theisen washers in the secondary washing plant. Fig. 3 shows dia-

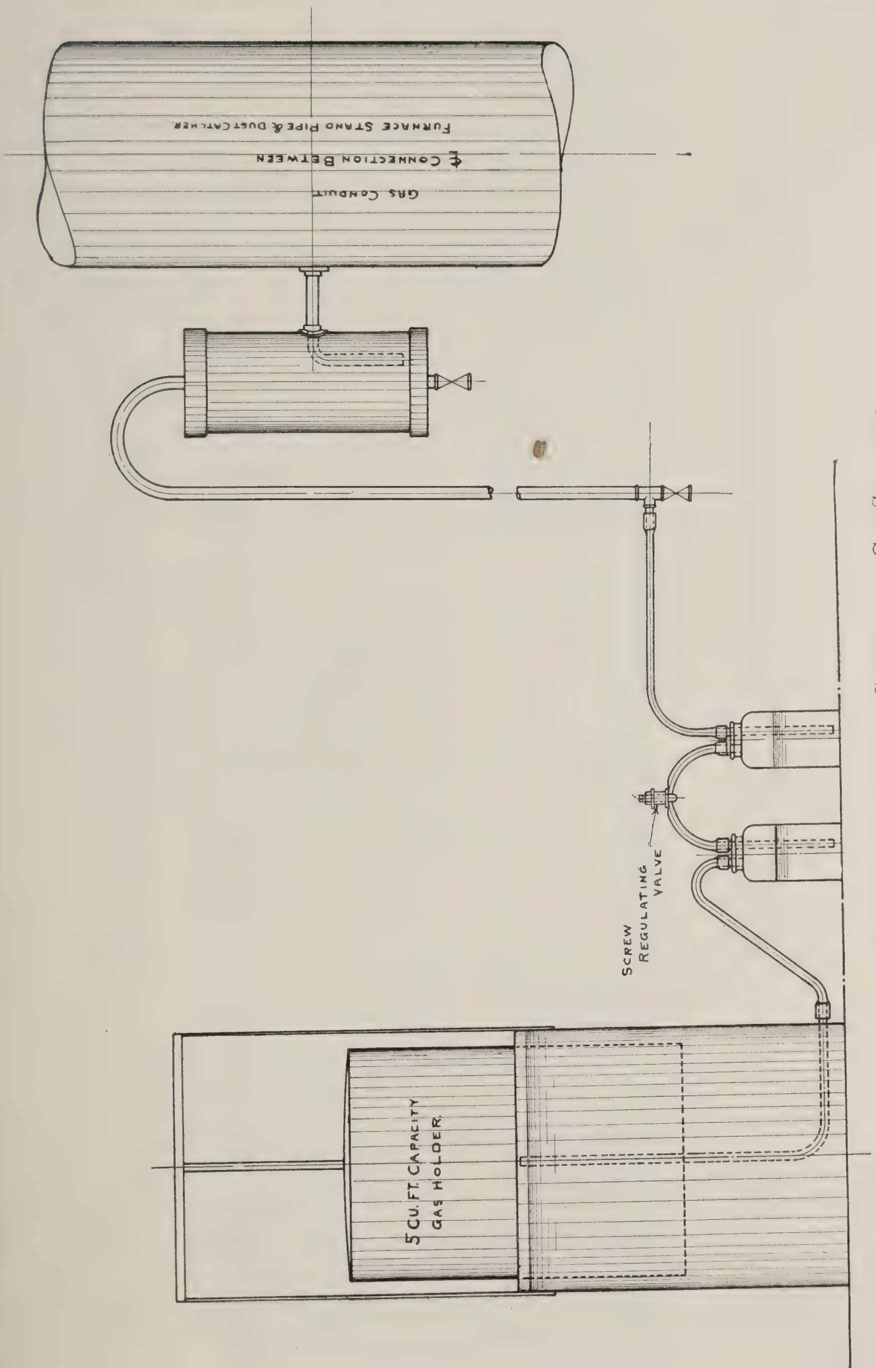


FIG. 5 APPARATUS FOR CONTINUOUS GAS SAMPLING

DAILY CHEMICAL REPORT For 24 Hours Ending 6 A.M. Tuesday, Sept., 21st

BLAST FURNACE GAS
Samples taken at Gas power station Laboratory

TIME	CO 2	CO	H 2	CH 4	$\frac{CO_2}{CO}$	$\frac{CO}{CO_2}$	B.T.U. Per Cubic Foot By Analysis	B.T.U. Per Cubic Foot By Junkers' Calorimeter
	CHEMICAL ANALYSIS % By Volume							
9:00 A.M.	8.1	31.1	3.6	.2	.260	3.84	112.6	112.0
9:30								111.9
10:00								111.0
10:30								121.2
11:00	8.2	32.2	3.7	.2	.255	3.92	116.5	119.9
11:30								116.2
12:00 NOON	8.2	32.2	3.0	.1	.255	3.92	113.6	114.0
12:30 P.M.								113.1
1:00								112.5
1:30								111.1
2:00								110.7
2:30								113.1
3:00								112.5
3:30								112.7
4:00	8.7	31.0	5.0	.1	.280	3.56	115.2	114.3
4:30								114.7
AVERAGE	8.3	31.6	3.8	.2	.262	3.81	114.5	113.8

		DAY TURN	NIGHT TURN	AVERAGE
Weather		Fair	Clear	—
Barometer		29.40″	29.31″	29.35″
Direction of Wind		Southwest		
Test Started		8:40 A.M.	5:00 P.M.	—
Test Ended		4:40 P.M.	8:00 A.M.	—
Number of cubic feet of gas passed through filter	Before Scrubbers	39.10	54.20	46.65
	After Scrubbers	59.22	100.63	79.92
	After Theisens	—	—	—
	A.C. Station Gas	88.36	148.37	118.37
	Blowing Eng. Gas	—	—	—
Moisture Grains per cu. foot	Before Scrubbers	—	—	—
	After Scrubbers	14.719	—	—
	After Theisens	—	—	—
	A.C. Station Gas	8.984	—	—
	Blowing Eng. Gas	—	—	—
	In Air	6.49	6.16	6.32
Flue Dust Grains per cu. foot	Before Scrubbers	3.8830	4.0812	3.9821
	After Scrubbers	.8147	.6884	.7516
	After Theisens	—	—	—
	In A.C. Engine Gas	.0085	.0052	.0068
	In Blowing Eng. Gas	—	—	—
% Dust Removed-Scrubbers		79.0	83.1	81.0
% Dust Removed-Theisens		98.9	99.2	99.0

Note: All figures given on this report are reduced to standard conditions i.e.
Temperature 62°F, Barometer 30″ Mercury
The heat value Constants are;
CO---324.45 B.T.U. PER CUBIC FOOT
H2---274.78 " " " "
CH4--915.08 " " " "

Metric Conversion Factor
1 Gram-15.43 Grains
MacFarland's tables of Reduction
Factors are used for all calculations within their scope.

OBSERVATIONS-
Furnace 1 on Ferrosilicon.
" 2 Basic iron.
" 3 " "
" 4 " "
" 5 Bessemer iron.
" 6 " "

3 Gas engines in operation.

Wm. Brady. ———— CHIEF CHEMIST
Peterson.

FIG. 6 FORM FOR DAILY CHEMICAL REPORT

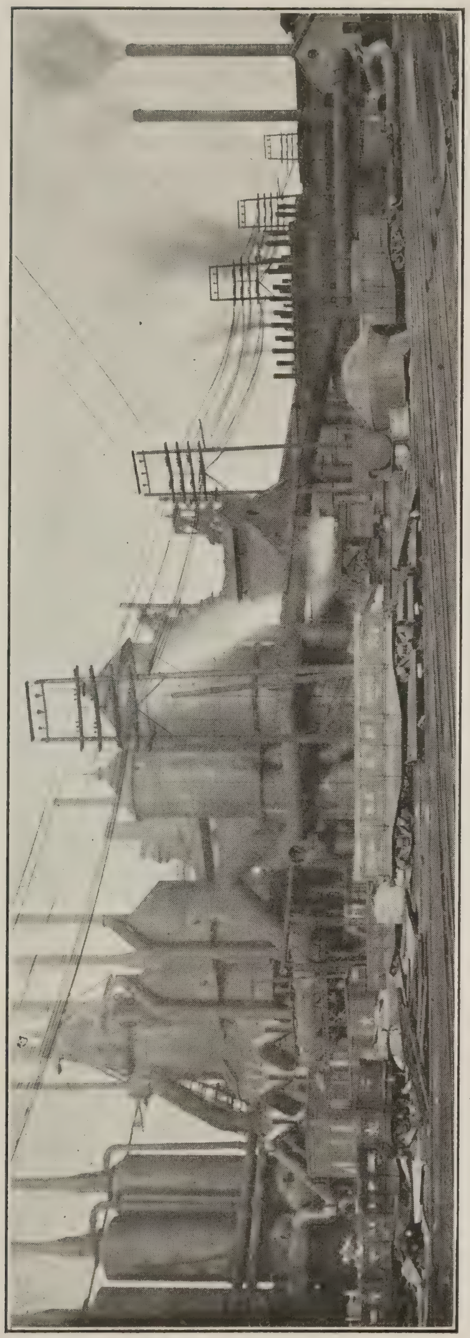


FIG. 7 GENERAL VIEW OF GAS-CLEANING PLANT, 1908

grammatically the general arrangement of the complete cleaning plant as it existed in 1908, while Fig. 7 gives a photographic view. When the gas-cleaning plant was designed in 1906, the raw gas was not cleaned except by the usual small dry dust catchers at the end of the downcomers of each furnace, and it was decided to install two special dry dust catchers of large capacity to remove the bulk of the heavy dust.

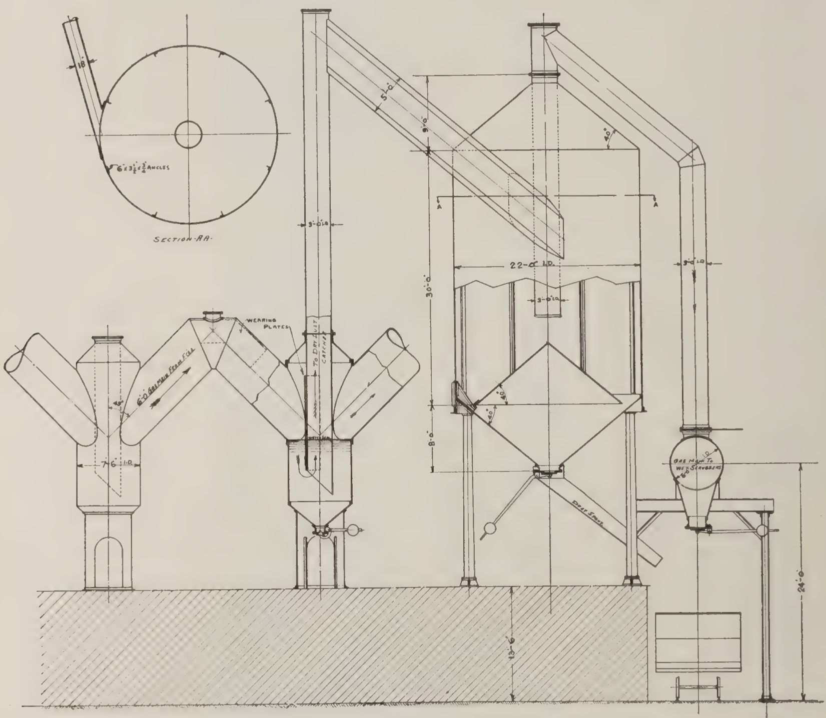


Fig. 8 Dry Dust Catcher

PRELIMINARY GAS-CLEANING PLANT

36 The raw gas on leaving the overhead gas flue first passes a water seal serving to shut off the gas power plant from the general system in case of necessity, and enters an unlined self-cleaning zigzag gas flue 6 ft. in diameter. Fig. 8 shows in detail the plan, and Fig. 9 a photograph of the zigzag flue and the dry dust catchers. It was originally intended to increase the capacity of the preliminary cleaning plant subsequently, by the addition of enough dry and wet scrubbers to clean the total quantity of gas produced by all six furnaces, for use

under boilers and in hot blast stoves, and provision for these additions was made when designing the cleaning plant. By means of water seals in the "dust legs" supporting the zigzag flue, and spectacle valves at the points of discharge of the dry-dust catchers into the "collecting main," each dust catcher may be shut off during the operation of the plant, in case of repairs or cleaning. As seen in the illustrations these water seals were designed on the principle of the Crawford valve, and by cutting off the ends of the inside pipes at an angle, it was intended to provide means of regulating the amount of gas passing through each dry dust catcher by filling the seal with water to a certain height, which was regulated by telescopic overflow.

FIG. 9 ZIGZAG MAIN AND DRY DUST CATCHER

37 Neither zigzag main nor dry dust catchers were lined with fire brick, as had heretofore been the practice with all gas pipes conveying raw gas of high temperature, as it was desired to take advantage of the reduction of temperature by radiation of heat through the unlined plate work. The results given elsewhere prove that the desired object was very satisfactorily accomplished. At all points of sudden change in direction of the flow of gas, "wearing" plates were provided, as excessive wear of the plate work, from the impinging of the dust-laden gas, was expected. These plates can be removed and replaced by new ones, through manholes arranged for this purpose. The dust

legs, as well as the dry dust catchers, were raised above the yard level high enough to permit the disposition by gravity of the accumulated flue dust into railroad cars by means of bell valves and dust spouts.

38 The dry dust catchers, two in number and operating in parallel, were 22 ft. in diameter by 31 ft. high with 9-ft. cones at each end. The choice of the diameter of the dry dust catchers, as well as of the

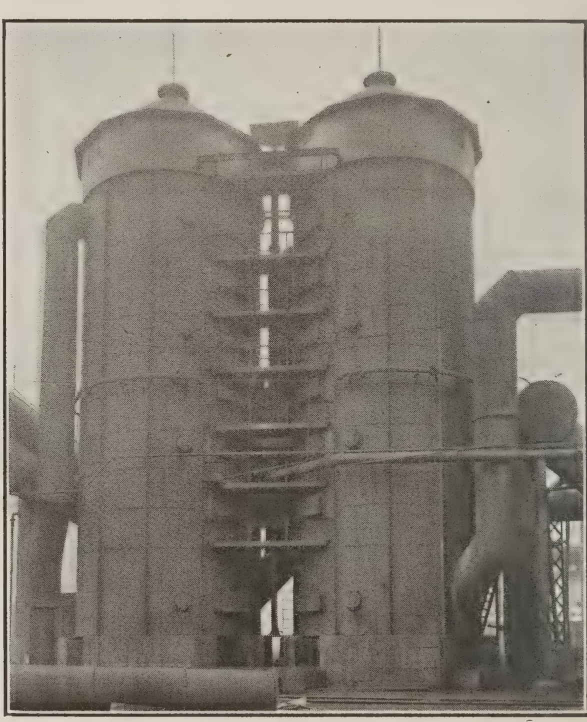


Fig. 10 Wet Scrubbers

wet scrubbers, was accidental, as it happened that some old 22-ft. hot blast stove shells were available at the time. The dry dust catchers were given tangential gas inlets, assuming that by this arrangement, and by inclining the flattened gas-inlet pipe, the gas would be caused to travel in long spirals from top to bottom, thus lengthening the path of the gas, and angle irons were placed vertically on the inside of the shell to provide for increased friction while the gas was traveling through the dust catchers at the slow rate of 1.5 ft. per sec. The

bottom cone was separated from the cylindrical part of the dust catcher by an inverted cone arranged umbrella-wise to prevent the dust accumulated in the bottom cone from being stirred up by disturbances caused by furnace slipping and re-entering the gas. The gas left the dry dust catchers near the apex of the umbrella, passing through a self-cleaning pipe into the collecting main and hence to the wet scrubbers. Explosion doors were arranged for on the dry dust catchers, but were eventually bolted down as unnecessary.

39 From the 6-ft. 6-in. collecting main, the dry-cleaned gas passes to the wet scrubber, shown in Figs. 10, 11*a*, 11*b* and 11*c*, through self-cleaning pipe lines. The piping arrangement permits the operation of the original two wet scrubbers in series, or in parallel, by turning a num-

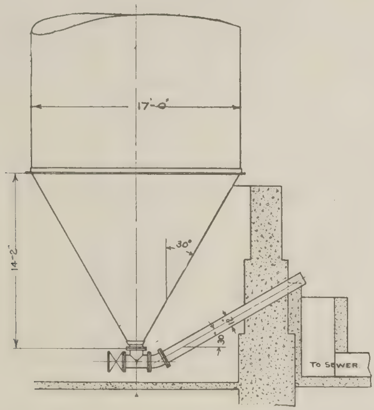


Fig. 11*a* Detail of Scrubbers Nos. 1 and 2

ber of spectacle valves, and water seals allow the shutting-off of either wet scrubber for cleaning, without interfering with the operation of the power plant. From the start the two wet scrubbers were operated in series, the total quantity of gas consumed by the engines passing first one and then the other. The gas enters each wet scrubber at the bottom of the shell, which is 22 ft. in diameter and 55 ft. in height. The inside is divided horizontally into six compartments, each containing eight rows of slats made of clear No. 1 white pine dressed all over. Each system of slats is supported independently by I-beams and angle irons riveted to the shell. The slats are 5 in. high and $\frac{7}{8}$ in. thick, by about 5 ft. 6 in. long, and ten slats on an average are nailed to distance pieces, forming hurdles about 5 ft. 6 in. long, and 3 ft. 7 in. wide, a size and weight which permit of easy handling. An

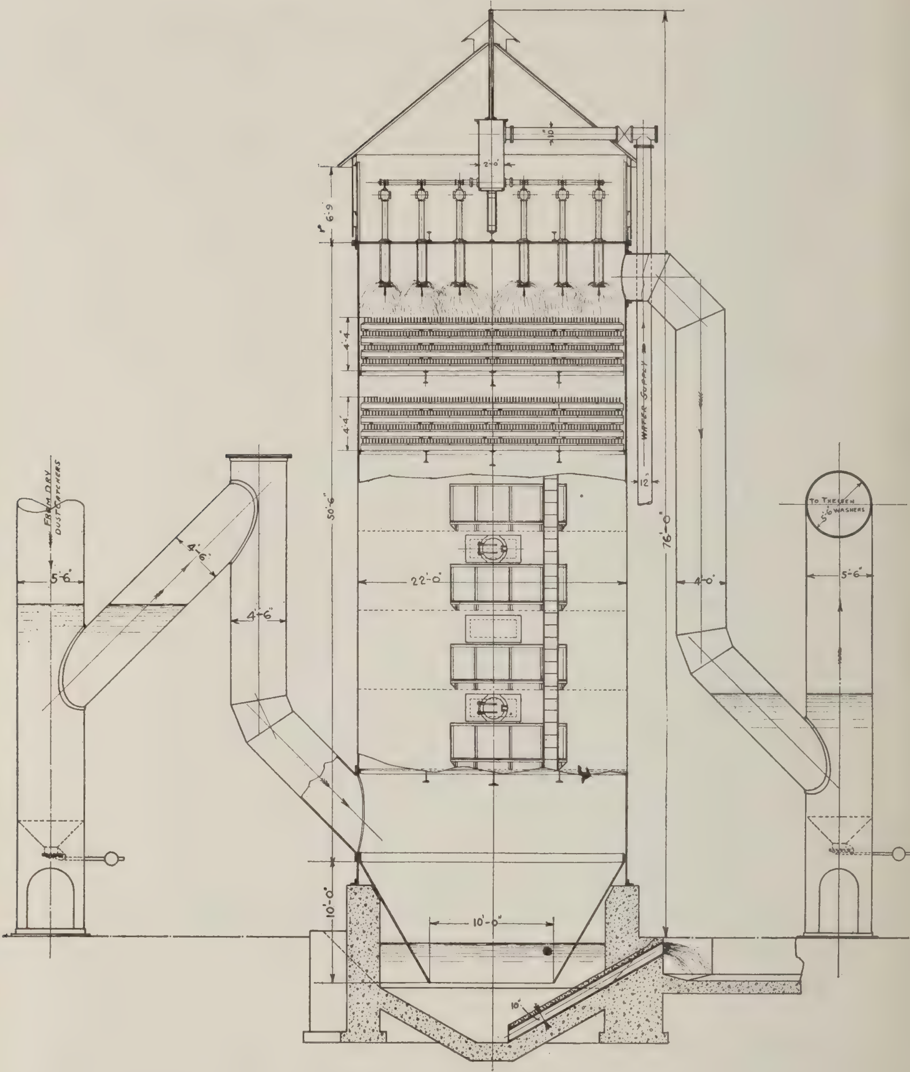


Fig. 11*b* Section of Scrubbers Nos. 3 and 4

interior view of the scrubbers with the hurdle arrangement is shown in Fig. 12.

40 Profiting by the experience gained elsewhere with wet scrubbers of the same kind, flue dust bridging over between slats and clogging the hurdles, it was decided to space the slats in the following manner;

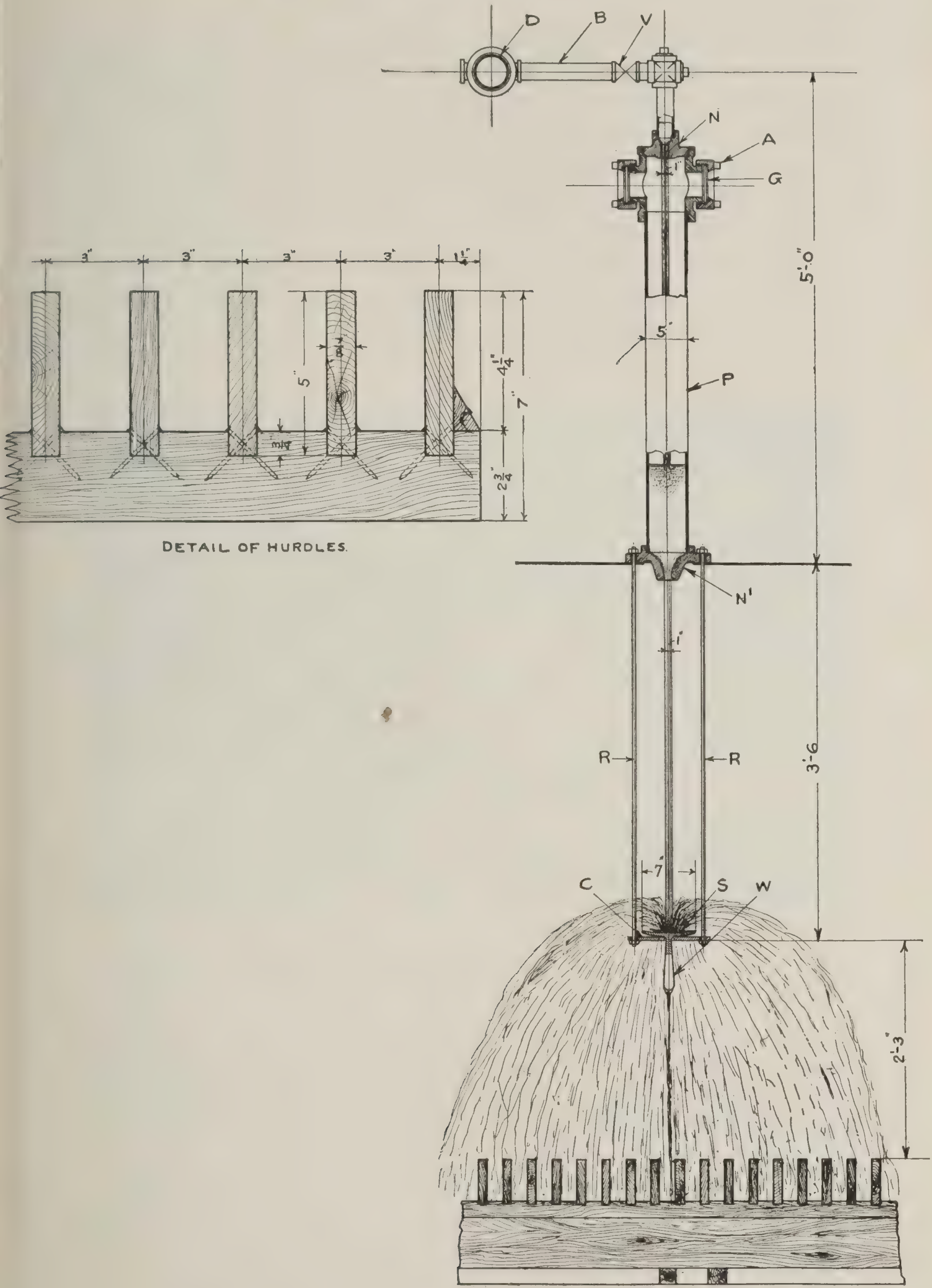


FIG. 11c DETAIL OF SPRINKLERS AND HURDLES

In the three lower compartments of scrubber No. 1, which receives dry-cleaned gas containing a large amount of coarse flue dust, the average distance of the slats was made 9 in. while in the three upper compartments the slats have 6 in. spacing. In the three lower compartments of scrubber No. 2, receiving gas already washed in the first scrubber, the slats were spaced with $4\frac{1}{4}$ in. centers, while the corresponding distance is 3 in. in the three upper compartments of the second scrubber. All hurdles were placed in the different rows and compartments in such a way that the slats of each upper row straddle the slats of the row immediately below, thus obtaining a continuous

Fig. 12 Interior View of Scrubbers Showing Hurdles

checker arrangement without any channels. A space of about two feet was left between each two hurdle sets, making each compartment accessible for cleaning or changing hurdles without removing the whole filling, and manholes and platforms were provided to facilitate this work.

41 The top of the wet scrubbers is formed by $\frac{5}{16}$ in. flat covers, supported by 8-in. I-beams. Each cover plate supports 36 sprinklers, shown in Fig. 13 and in detail in Fig. 11*c*, distributed over the entire section. Each sprinkler consists of two nozzles N and N', and a cast-iron spray plate S with slightly curved surface and weight W

to insure horizontal position. The spray plate is inserted in the center hole of the crosspiece C, which is supported by two rods R fastened to the top cover plate. The upper nozzle of 1 in. diameter is mounted on one branch of a cast-iron cross, while the opposite branch is connected by a 5-in. wrought-iron pipe P about three feet in length, to the lower nozzle N' of equal size. The other two branches of each cross are closed by caps A, containing plate glass discs G, $\frac{1}{4}$ in. thick and 4 in. in diameter. The open water tank located above the wet scrubber supplies the washing water under a small but constant head, through distributing pipes D and branch connections B to the sprinklers, the amount of water being regulated by valves V. The operation of these sprinklers is obvious. A stream of water falls in each sprinkler through a distance of about eight feet, breaking up into an exceedingly fine mist by impinging on the spray plates, and as the sprays of the 36 sprinklers overlap each other, the distribution of water is perfect.

42 These sprinklers were tested before the wet scrubbers were put into operation, and one minute after turning on the water there was not a dry spot on the inside of the scrubbers. Originally these sprinklers were without the nozzle N'. The action was the same as far as the distribution and the atomization of the water was concerned, but with the serious drawback that the dirty gas could reach the upper part of the sprinklers and deposit dirt on the sight glasses, which soon became useless. By the insertion of nozzle N' this trouble was successfully overcome, as the water flowing through this nozzle completely seals the upper part of the sprinklers so that the glasses can be removed during operation without danger from escaping gas. The great advantages of this type of sprinkler are that a clogging of the water passages can never occur, and uniform distribution of the water can always be obtained. Their operation has been exceedingly satisfactory. All water piping on top of the wet scrubbers is housed-in for protection against frost.

43 The lower part of the wet scrubbers dips with a conical extension into a water seal provided in the concrete foundation. The muddy water is carried off through an overflow pipe reaching to the bottom of the seal, thus keeping the water in constant circulation and thereby effectively preventing any accumulation of mud in the seal. It was found advantageous, however, to introduce into this overflow passage and reaching to the bottom of the seal, a 1-in. pipe through which water under pressure is constantly discharged, stirring up the

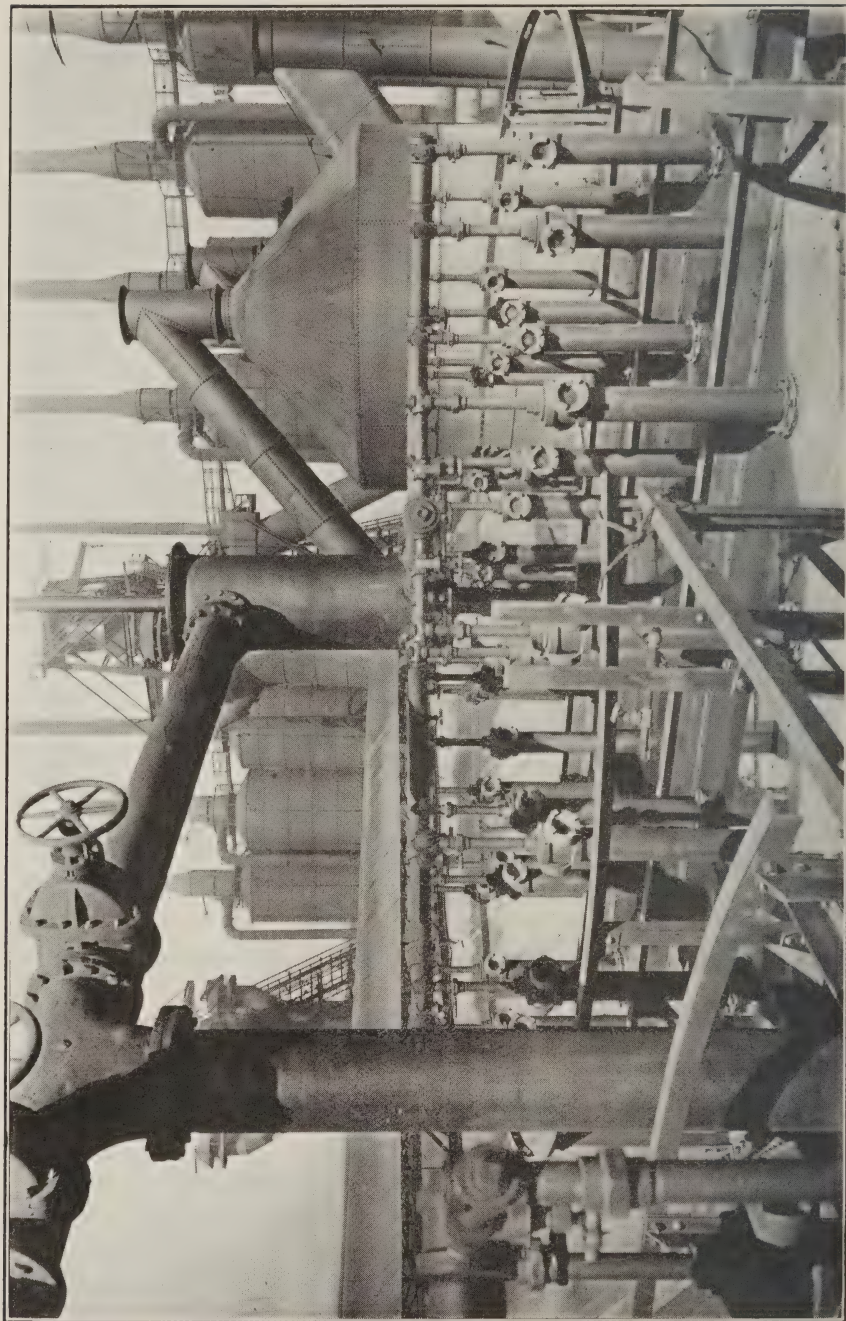


Fig. 13 Water Sprinklers

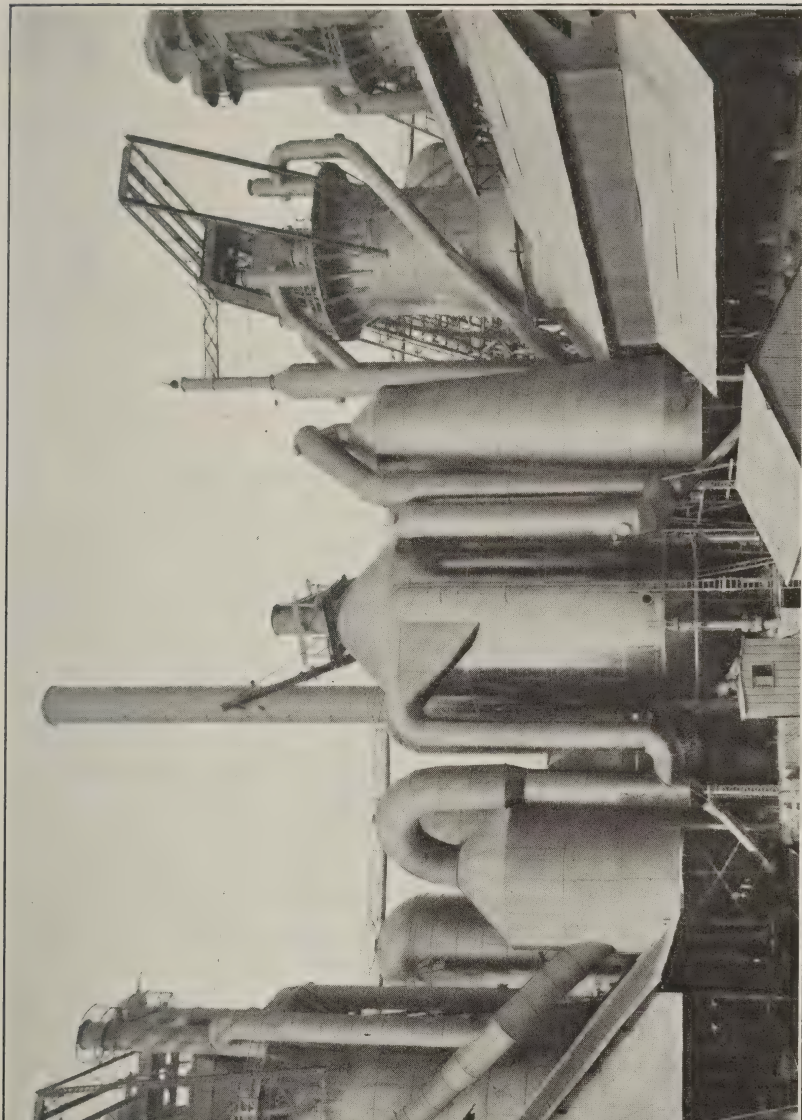


FIG. 14 DRY DUST-CATCHER SYSTEM AT FURNACES 5 AND 6

sediment and keeping the water in motion. On the two new wet scrubbers recently installed this type of water seal was abandoned in favor of the simple arrangement shown in Fig. 11*b*.

44 The waste water from these scrubbers flows into a large settling tank, which was installed to prevent a possible clogging of the main sewer. Both compartments of the settling tank, which was in operation about two years, have since been completely filled with mud, but as the dust in suspension does not prove to be troublesome in the sewers, little attention is being paid to their regular cleaning.

45 When in the early part of 1909 four additional gas blowing-engines for blast furnaces No. 1 to 4 were purchased, an increase in the capacity of the gas-cleaning plant was necessary, and since in the meantime the question of dry cleaning the raw gas at the furnaces had been solved, it was decided to change the two original dry dust catchers into wet scrubbers. As soon as the voluminous dry dust catcher system at the furnaces (Fig. 14) was in operation its effect was noticed in the materially reduced efficiency of the dry dust catchers in the preliminary gas-cleaning plant. These had formerly removed a great deal of heavy, dry flue dust, but suddenly became practically useless, and only a little dust, now in the form of mud, was taken out. While the change from dry dust catchers to wet scrubbers was being made, in the second half of 1909, only the two original wet scrubbers were in operation. In the near future four wet scrubbers, of sufficient capacity to take care of the preliminary washing of the gas required by 40,000 h.p. in gas engines, will be in use. Fig. 15 shows the general arrangement of the gas power installations at present. Fig. 16 is a diagram of the path of the gas through the new wet scrubbing plant, showing the combinations in which the four scrubbers can be operated. Fig. 7 shows the gas main carrying the clean gas from the wet scrubbers to the secondary cleaning plant. Attention is called to the design of the supports of this pipe line, which are built as so-called dust legs, wherein water and flue dust are deposited, and drawn off occasionally through bell valves. The clean gas main, while not self-cleaning, is arranged to slope in both directions. At certain intervals circular water pipes with spray arrangements are installed for flushing the clean gas main, and at the points where a sudden change of direction of the gas occurs sealed holes are provided, through which a thorough cleaning of the pipe line can be made, with fire hose and high-pressure water, during the operation of the plant.

46 After leaving the wet scrubbers, the clean gas, as it is called, is in such a condition that it could be used under boilers and in hot blast

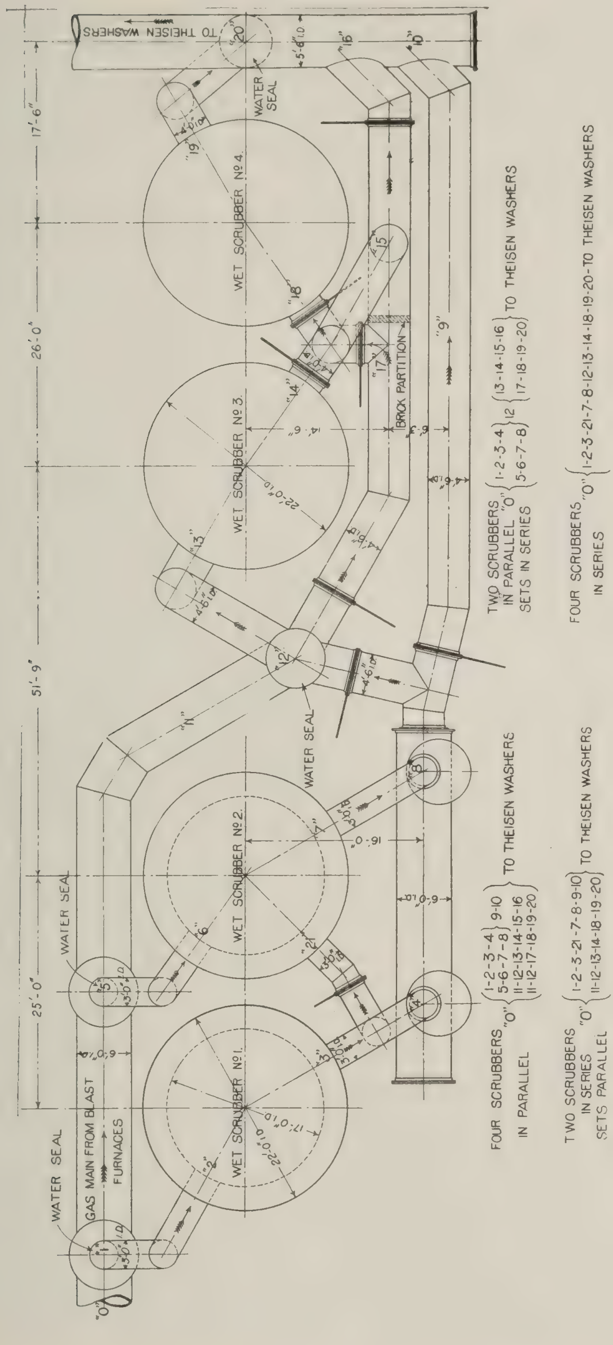


Fig. 16 Flow of Gas Through Wet Scrubbers

stoves, if desired. In 1909 it contained an average of not more than 0.318 grains of dust and not over 5.62 grains of moisture per cubic foot. Its purity therefore nearly complied with the requirements of European blast furnace plants where 0.5 grammes per cubic meter (0.218 grains per cu. ft.) is considered a desirable degree of cleanliness for stove and boiler gas.

IMPORTANCE OF GAS-CLEANING

47 Clean gas as delivered by the preliminary washing plant is, however, not sufficiently purified for gas engine purposes. Not so very many years ago it was thought in good faith that gas engines could operate on dirty gas, and it required years of costly experimenting to develop the art of gas purification to its present perfection, after the far-reaching importance of the problem was at last understood. Its magnitude can be better appreciated if the total quantities of gas and dust which are handled in such a cleaning plant during a certain period of time are considered. The following figures apply to the gas power plant under discussion for the year 1909.

48 The total number of kilowatt-hours generated by the gas engine installation was 50,494,100. The average heat consumption per kilowatt-hour was 17,234 B.t.u. The average heat value of the gas by calorimeter was 98.3 B.t.u. per cu. ft. The gas engines consumed therefore per kilowatt-hour 175.3 cu. ft. of gas, or in the year 1909 a total of 8,851,615,730, or nearly 9,000,000,000 cu. ft. This total quantity of gas reached the wet scrubbing plant containing on an average 1.533 grains of dust per cubic foot. There were consequently carried into the wet scrubbers during the whole year 1,938,500 lb. or 865 gross tons, of flue dust. To appreciate fully the meaning of this enormous figure it may be remembered that to haul this quantity away, a freight train of twenty gondola cars of 100,000 lb. capacity would be required. The average amount of dust in the clean gas for the year was 0.3183 gr. per cu. ft.; so that it carried 402,500 lb., or 180 gross tons of flue dust into the secondary cleaning plant. The difference of 685 tons was taken out by the wet scrubbers and carried off into the settling tanks. Expressed in per cent of the original quantity of dust, the wet scrubbers removed 80 per cent of the impurities. The Theisen gas washers further took out from the gas 176.7 tons, leaving only 3.3 tons in the fine gas, since the average amount of dust in the latter was 0.00583 grains per cu. ft. The Theisen washers had therefore an efficiency of 98 per cent, shared by clean gas main, fine gas main and gas holder.

The over-all efficiency of wet scrubbing and secondary cleaning plants was 99.5 per cent, since of the original 865 tons 861.7 tons was removed from the gas and only 3.3 tons entered the gas engines. Of the latter quantity only a small amount remained in the engine cylinders, since the bulk of the dust is swept into the atmosphere at each exhaust stroke. These figures will give a good idea of what it would mean if gas engines were operated on clean gas, not to speak of dry-cleaned gas, and yet this was attempted in the early history of the blast furnace gas engine.

SECONDARY CLEANING PLANT

49 It is generally recognized that blast furnace gas cannot be cleaned sufficiently for engine purposes without the expenditure of power, and that a satisfactory refining can only be performed in rotary gas washers on the "dynamic" principle, in contradistinction to the preliminary washing for which "static" methods are usually found to be fully adequate. Among the rotary gas washer systems on the market, the Theisen washer is considered to be mechanically well designed and very efficient. The Theisen washer installation consisted in 1909 of four washers, each of 15,000 cu. ft. per min. capacity. One additional washer has been installed recently on account of the new gas blowing-engines. Fig. 17 is an interior view of the Theisen washer building, and Fig. 18 shows the plan and elevation of this installation. The Theisen washers are arranged in two rows in a fireproof building, with the clean gas main overhead between them, and inlet pipes to the suction end of each washer. The outlet pipes pass through the building to water separators and to a ring main which delivers the gas through a 5 ft. fine gas main about 500 ft. long to the power station gas holder, and through a 4 ft. 6 in. fine gas main about 1,000 ft. long to the blowing-engine gas holder.

50 The Theisen washer, shown in sectional view in Fig. 19, consists essentially of a closed drum fitted on its outer surface with longitudinal blades arranged in spirals. This drum, supported by a shaft in two water-cooled ring-oiling bearings, rotates at high speed inside of a stationary casing of conical shape. The inlet end is equipped with suction vanes while on the discharge end an exhaust fan is firmly attached to the drum. The gas is introduced into the annular space between revolving drum and conical casing and discharged by the fan into a water separator. The operation of the washer is as follows:

51 The suction vanes draw the gas from the inlet pipe and deliver it to the longitudinal vanes, which have an inclination to the axis of

the drum so as to oppose the flow of the gas through the washer. The discharge fan at the outlet end of the drum, however, overcomes this tendency and discharges the gas under positive pressure of a few inches of water. The clearance between the outer edge of the longitudinal blades and the inner surface of the stationary casing is not more than 1 in. and the gas passing through this narrow space under high pressure imparts to the water introduced at several points into the casing a movement in long spirals in an opposite direction to its own travel. This flow of water in the form of a film covering the inner surface of the stationary casing, is assisted by the conical shape of the latter, taper-

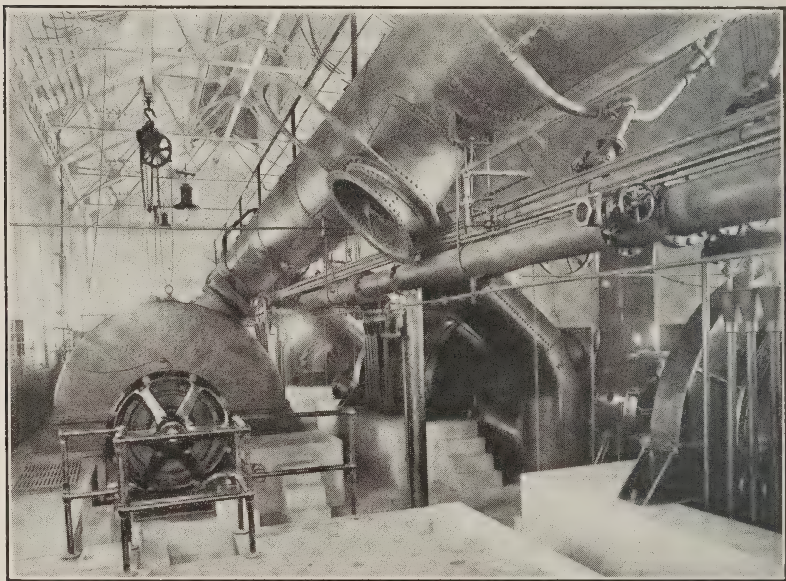
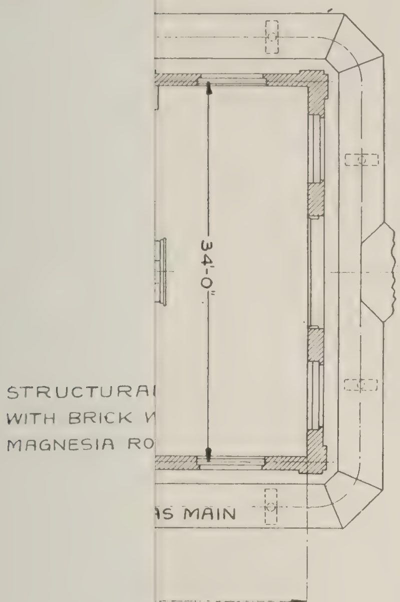
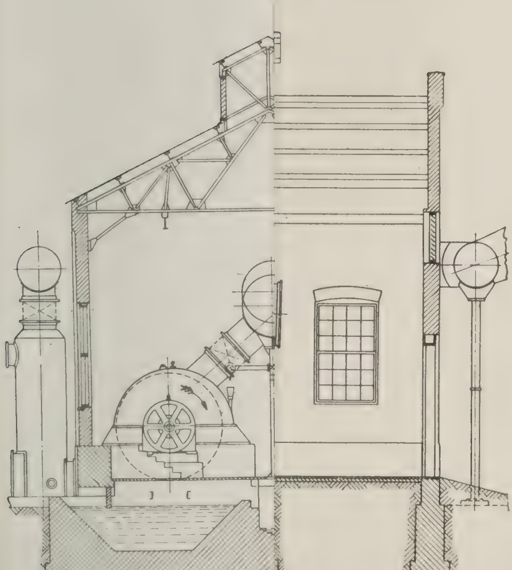


Fig. 17 Interior View of Theisen Washer Building

ing off towards the gas inlet. The surface of contact between gas and water is materially increased by wire netting, closely fitting the inside of the casing. By the intimate action of the water on the gas the dust particles are thoroughly moistened, and being weighed down by water drops, are thrown by centrifugal force into the rotating film of water, to be carried away through a seal into the sewer. The gas leaving the washer is charged with more or less moisture in the form of mist, which is removed from the gas in the Theisen washer separator, consisting principally of a removable box filled with iron shavings held in place





SECTION

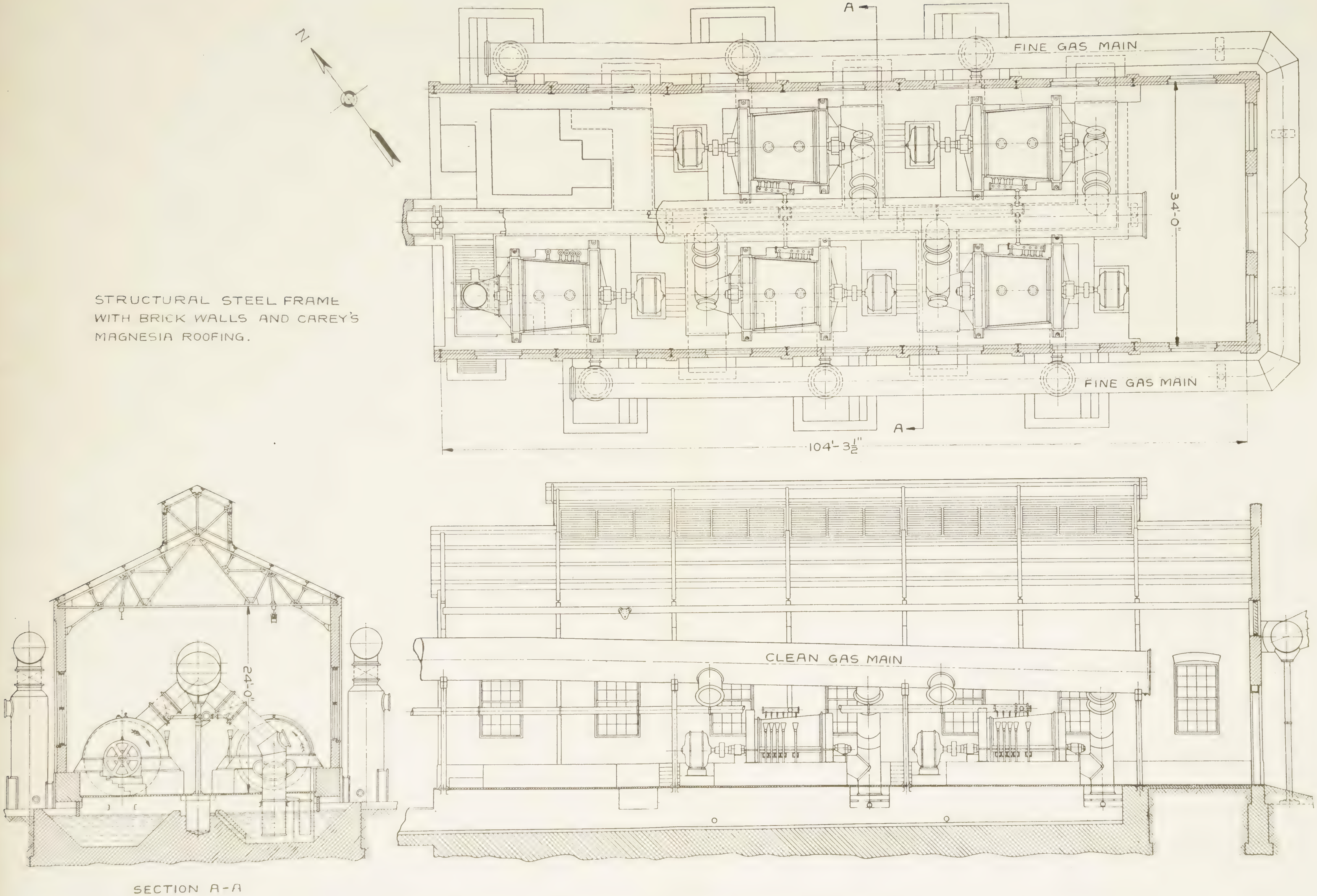
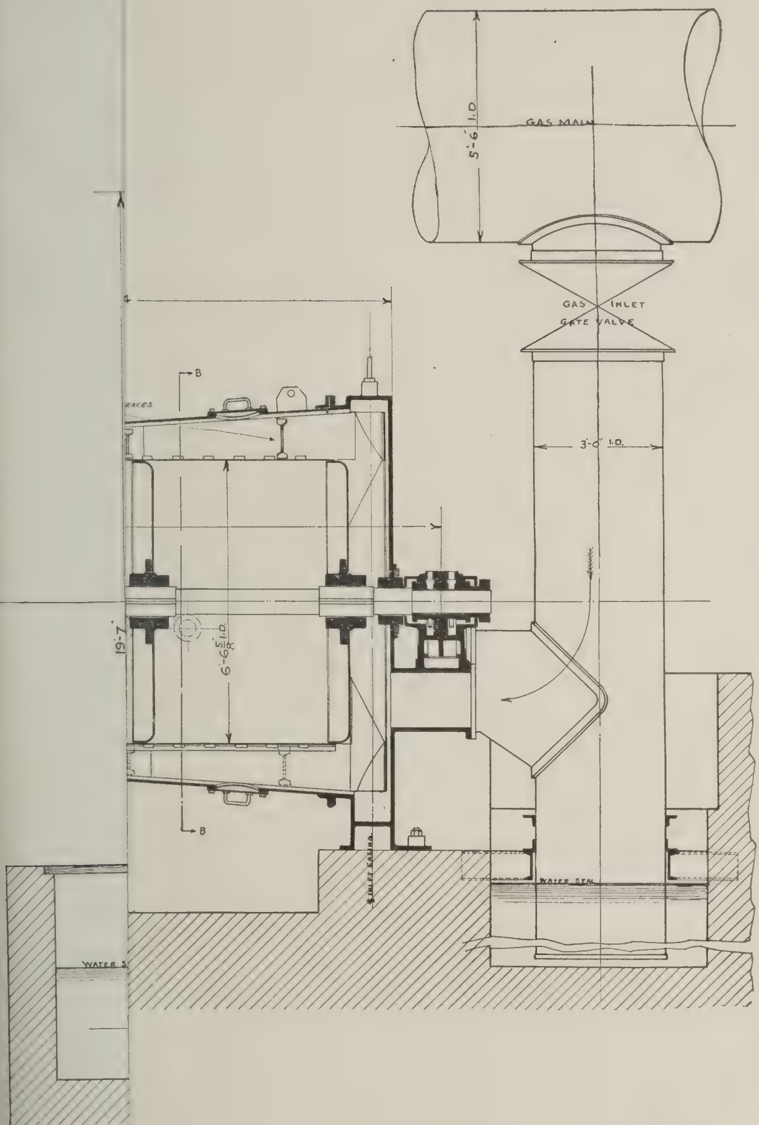


Fig. 18 Plan and Elevation of Theisen Washer Installation


5'-6" I.D.
GAS MAIN
GAS INLET
GATE VALVE
3'-0" I.D.
B
B
RACES
19'-7"
6'-6 5/8" I.D.
INLET KOERN
WATER SEAL
WATER S

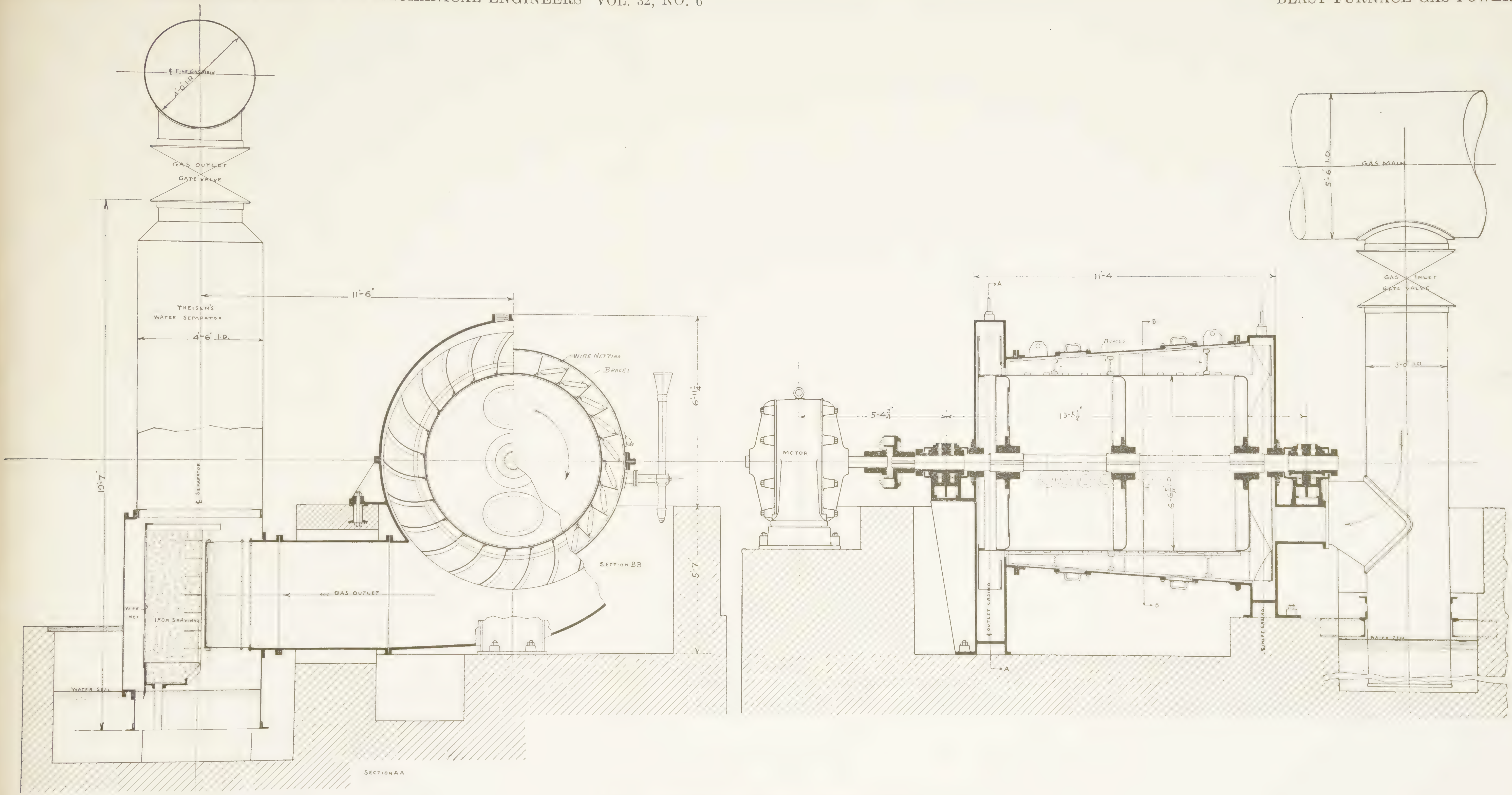


FIG. 19 SECTIONAL VIEW OF THEISEN WASHER

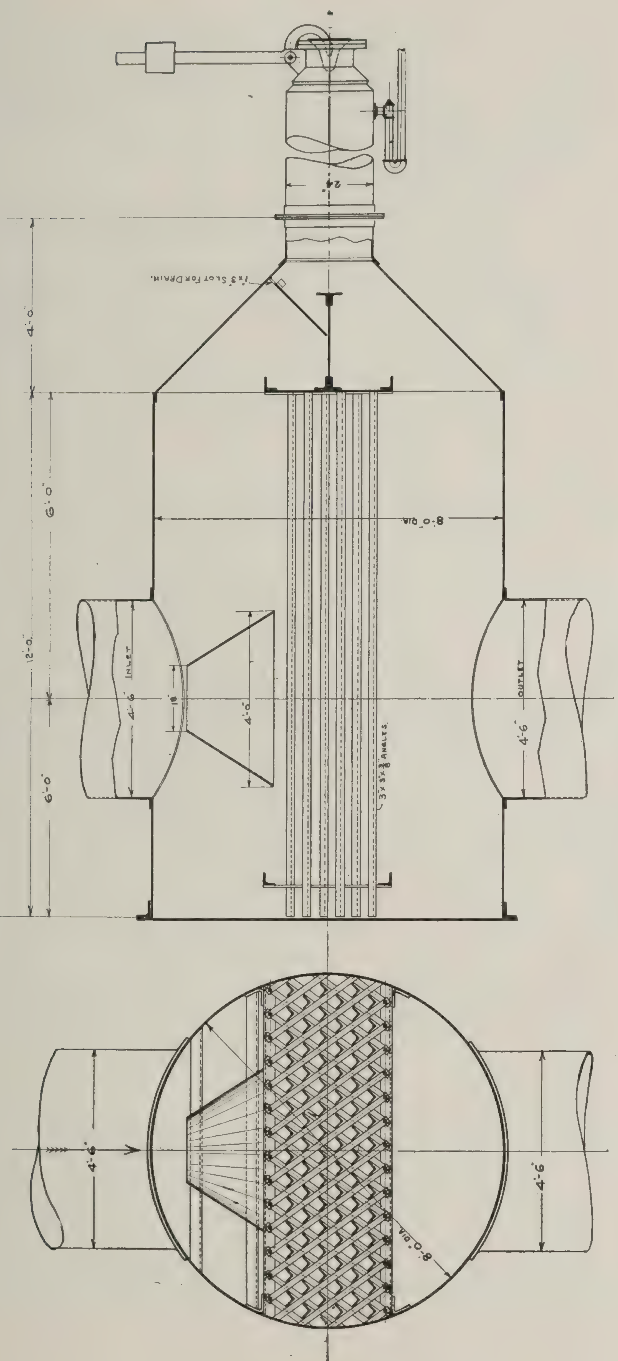


FIG. 20 WATER SEPARATOR

by wire netting. The gas striking the iron shavings with great velocity deposits its moisture, and as it has to reverse its direction it cannot pick it up again, but leaves the separator in a comparatively dry condition. Gate valves serve for regulating the quantity of gas entering the different washers, and for shutting off any washer without interfering with the operation of the plant.

52 The two gas holders, of 100,000 cu. ft. capacity each, were installed primarily to give a constant pressure of about four inches of water column at the gas engine throttle. Incidentally, however, they serve as reservoirs and as water separators. Since a gas holder was originally not contemplated for the gas blowing-engine installation an additional water separator, shown in Fig. 20, was provided in the fine gas main to the gas blowing engine house, the design of which is based on a well-known principle. A number of angle irons serve as baffles, dividing the gas into a number of streams which are forced to change their direction several times while passing through the rows of angle irons. The latter are placed "straddling" similar to the wooden slats in the wet scrubbers. A bell valve at the end of a long pipe serves to remove the accumulated water. Each gas holder is of the single-lift type, with bell 59 ft. 6 in. in diameter by 36 ft. high. Both holders have separate gas inlet and outlet pipes to obtain continuous circulation in the holder and prevent the pocketing of stale gas. While in the power station holder this idea was carried out to the extent of having inlet and outlet pipes at opposite ends of one diameter, the blowing-engine gas holder has these pipes side by side, but with the inlet turned a little to impart to the gas a rotating motion. Inlet and outlet pipes can be used as water seals to shut off the gas holder in case of necessity. To prevent the possible collapse of the gas holder bell, in case the supply of gas should be interrupted and a vacuum created underneath the holder bell, a disc valve supported by chains from the holder crown is located exactly above the mouth of the outlet pipe as shown in Fig. 21. When the holder bell descends until it rests on its landing beams, this valve will close the outlet opening, preventing a vacuum under the bell.

53 The discharge pressure of the Theisen washers is about 8 in. higher than the pressure on the suction side, and as the latter is quite variable, the former will also vary within considerable limits. It is of course possible to regulate the pressure in the fine gas main by means of the gate valves arranged in the Theisen outlets; but since these pressure variations are almost continuous, the gate valves would have to be adjusted by the operators practically all the time, unless it

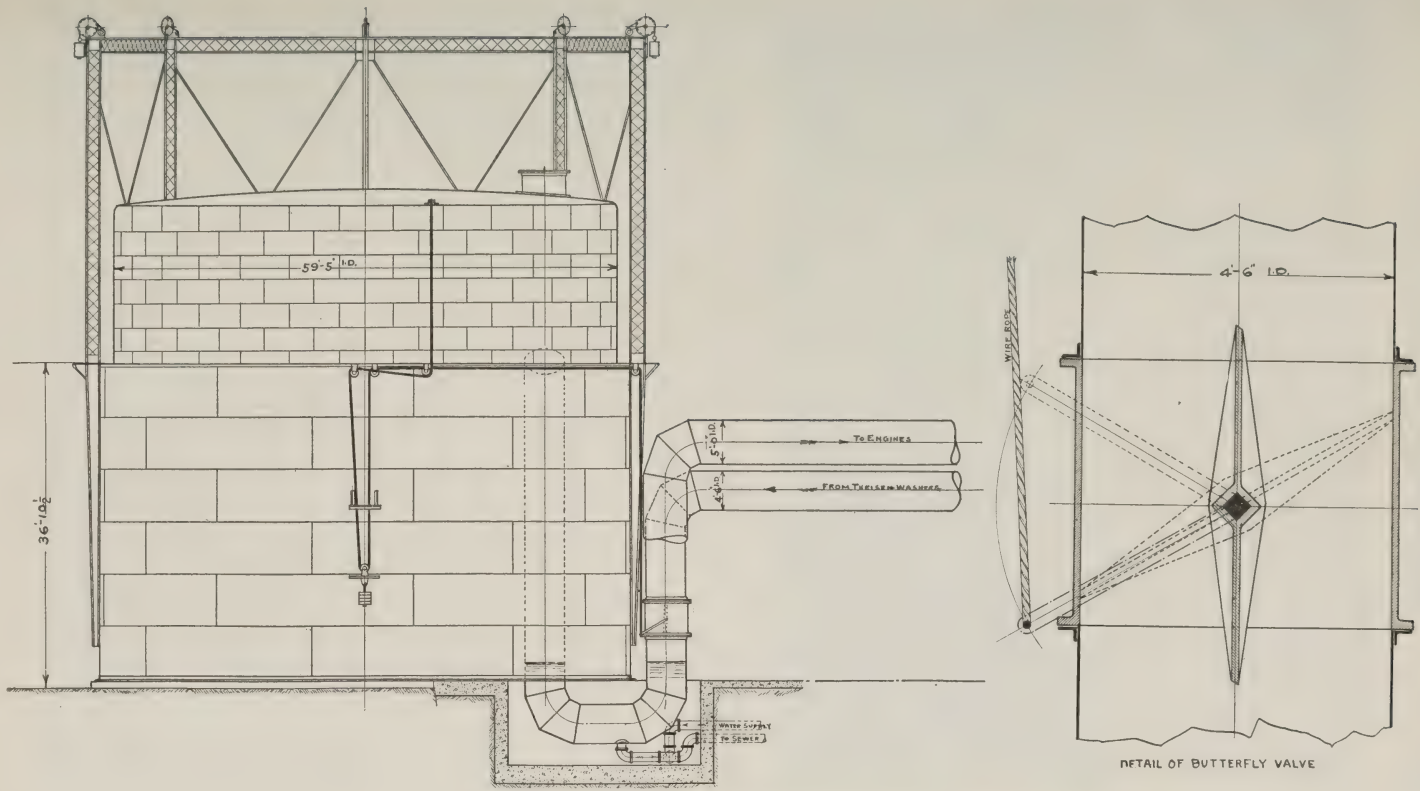


Fig. 21 Gas Holder and Automatic Butterfly Valve

were desired to resort to the complication of electrically operated valves under automatic control of the gas pressure. To simplify this necessary regulation a butterfly valve was installed in the inlet pipe to each gas holder and operated by the holder bell itself by means of a wire cable fastened to it and carried over a system of pulleys as shown in Fig. 21. This device works in the following manner:

54 When the gas holder is empty, the butterfly valve is wide open and the counterweight hanger *H* with weights *W* is in its bottom position. When the bell ascends, hanger *H* rises without affecting the butterfly valve until bumper bracket *B* is reached, which prevents the further travel of the hanger. The movable pulley *P* now becomes stationary, and the rising holder bell acts through cable *C* on the butterfly valve, throttling and finally tightly closing it, the effect being precisely the same as if the gate valves on the Theisen washer outlets had been throttled or closed. The washers continue to operate, but cease to deliver until the descending gas holder bell again opens the butterfly valve. This action is perfectly automatic and it is impossible for more gas to enter the gas holder than is being taken out, so that any number of gas engines can be started or shut down at any time without the slightest adjustment at the Theisen washers.

55 Without this automatic regulator this is what would happen: The weight of the gas container gives a constant pressure of 4 in. of water column in the outlet pipe and the Theisen washers deliver a constant quantity of gas as long as the gas pressure on the suction side is constant. With a certain number of gas engines in operation, and the gas demand equal to the gas supply, the gas holder bell will float in a certain position. If, however, one or several engines are stopped, the gas demand will decrease and as the gas supply remains constant the bell will rise into its top position, determined by the height of the water seal in the holder tank. Any further rise will break this seal, causing gas to escape from underneath the holder bell. This will continue until more engines are started and the gas demand is again equal to the gas supply, or until the Theisen outlets are sufficiently throttled to reduce the quantity of gas delivered. The disadvantages are obvious. Not only will the breaking of the seal cause large quantities of water to be thrown out, but the escaping gas, aside from being unnecessarily wasted, will dangerously foul the surrounding atmosphere. The automatic butterfly valve, balancing perfectly the gas demand and the gas supply, eliminates these troubles very effectively.

56 A by-pass line permits the operation of the gas engines directly if for any reason the holder is out of commission. Each holder de-

livers the gas into a large main located on the outside of the gas engine buildings, as shown in Fig. 22. Individual branch pipes lead the gas to each engine, which can be isolated from the gas main by bell valves or water seals, as shown in Fig. 23*a*. The gas pipes leading to the engines, as well as all gas mains, are carefully drained by automatic overflows, an important feature.

FIG. 22 EXTERIOR VIEW OF POWER HOUSE; GAS HOLDER AND GAS RECEIVERS

PERFORMANCE OF THE GAS-CLEANING PLANT

57 The physical qualities of the gas of importance from the standpoint of gas engine operation, are its pressure, temperature, dryness and cleanliness. These conditions, and particularly the last, if ascertained and suitably recorded at various stages of the cleaning process, are valuable indicators of the efficiency of the gas-cleaning plant; very few blast furnace plants, however, pay sufficient attention to their regular routine determination. As a rule tests are being made and results recorded only so long as the gas engine installation is new and therefore of all-absorbing interest. Particularly if the operation of the plant seems satisfactory, the interest is soon lost and the plant is left entirely to the care of the operating men, who soon are the only authorities on the machinery in their charge. The knowledge that can be obtained from them is of questionable value, as it is often based on good memory only, and gathered in hit-and-miss fashion. All operative results of a gas power plant should be recorded with as thorough care as is usually afforded the operation of steam plants, or even more, since the gas engine is more susceptible to variations in the quality of its fuel.

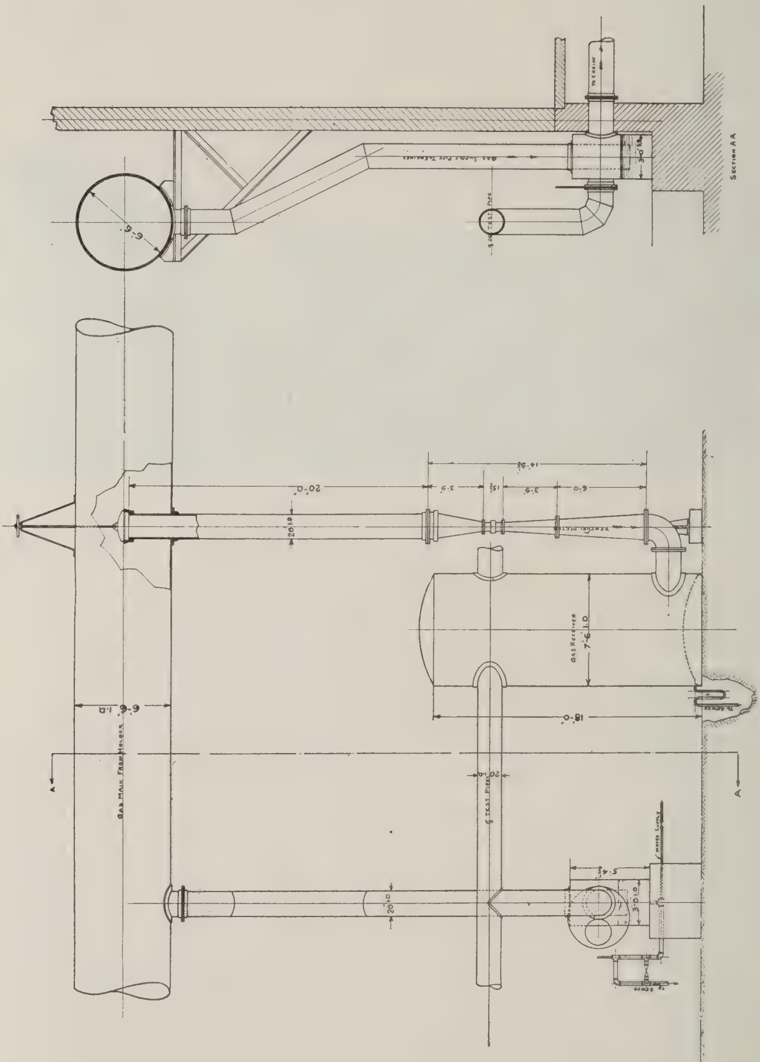


FIG. 23*a* GENERAL ARRANGEMENT OF TEST PIPING

58 The question is often asked, what can be the advantage of keeping exact records if the recorded results vary comparatively little during the year's operation, and whether the game is worth the candle, assuming that the expenditure is far in excess of the benefit derived. The questioner overlooks, however, that only by keeping such records can it be determined whether the conditions really are generally uniform; and that this uniformity is in many instances due to the careful watching and recording of the phenomena involved. Besides it was found, in over two years' experience at the plant under discussion, that the expense of "keeping the finger on the pulse" of the gas power plant is so small and so easily absorbed that it is insignificant. The expense connected with the maintenance of a special gas laboratory, for instance, has never as yet noticeably increased the cost of pig iron, and the three-hourly readings at the gas cleaning plant are being taken, without additional expense, by the operators themselves, who are assisted and checked in their work by recording instruments installed wherever expedient. It has been found, too, that the installation is being given much more care by the operators, since they are compelled to go over the whole plant on regular beats, in order to take the various readings. The general appearance of the gas cleaning plant shows unmistakably the influence of this continuous inspection. It is only natural that the operators should themselves become interested in their readings and compare the results from day to day. They soon make changes and improvements in the equipment, of their own accord, and will operate the plant at a much higher standard of efficiency.

59 Fig. 6 and Fig. 24 show two of the standard record forms, which are self-explanatory. The data collected on the various report sheets are tabulated and plotted on charts in the engineering department so as to show the daily, monthly and yearly averages. These records are very valuable from an operating point of view. The economy of a gas-purifying plant, for instance, is dependent on a number of elements, among which the plant efficiency is not of least importance. It involves the question of total cost per unit of gas, of electric light and power consumed in the plant, of operating labor, labor and material used in repairs, and lubricants in relation to the degree of cleanliness and the amount of moisture obtained by this total expenditure in the same gas unit. The majority of these elements can be controlled when the variations to which they are subjected are known, by voluntary or involuntary changes, but this knowledge can be acquired only through close and continuous observation.

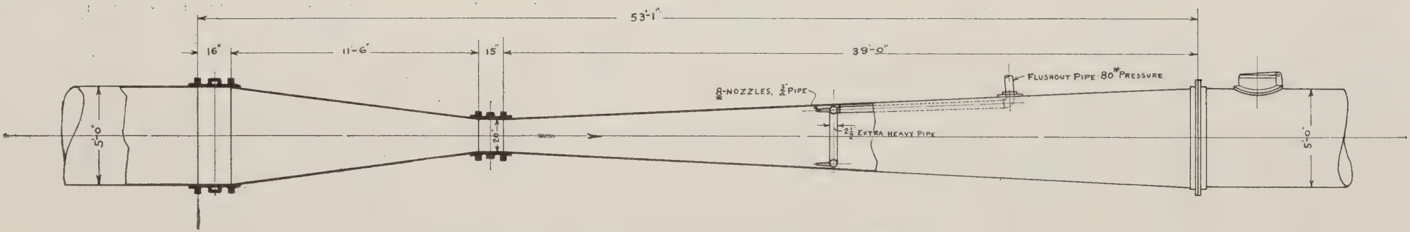


FIG. 23b TEST PIPING—60-IN. VENTURI METER

PRESSURES AND TEMPERATURES-DAILY REPORT FOR 24 HOURS ENDING 6 A.M. Thursday, October 7th, 1909.

TIME		GAS TEMPERATURES (°F) PRESSURES ("OF WATER) Entering Gas Cleaning Plant T	Entering Gas Cleaning Plant P	After Wet Scrubbers 1 T	After Wet Scrubbers 2 T	After Wet Scrubbers 3 T	After Wet Scrubbers 4 T	Before Theisen Washers T	Before Theisen Washers P	After Theisen Washers T	After Theisen Washers P	A. C. Sta. Gas Holder In T	A. C. Sta. Gas Holder Out T	B. Engine Gas Holder In T	B. Engine Gas Holder Out T	WATER TEMPERATURE Pump Suction	Wet Scrubber Waste 1	Wet Scrubber Waste 2	Wet Scrubber Waste 3	Wet Scrubber Waste 4	Waste Theisen Washers	Air Temperature	READING OF AMMETER ON THEISEN WASHER MOTORS AMPERES PER TERMINAL 1	2	3	4	5	6	7	8	WATER CONSUMPTION GAL./MIN. Scrub.	Th'sn	No. of Electric Engines in Operation	No. of Blowing Engines in Operation
DAY TURN	8 A.M.	218	17.0	—	—	77	64	68	16	68	22	67	65	—	—	67	—	—	96	69	70	66	—	—	155	155	—	—	—	—	1400	300	3	—
	11	196	16.5	—	—	83	74	76	16	76	21	75	74	—	—	69	—	—	98	71	72	76	—	—	155	155	—	—	—	—	1400	300	3	—
	2 P.M.	204	14.0	—	—	82	75	77	13	77	19	79	77	—	—	74	—	—	104	76	77	72	—	—	155	155	—	—	—	—	1400	300	3	—
	5	208	15.5	—	—	82	72	74	15	74	21	74	72	—	—	74	—	—	104	76	77	70	—	—	155	155	—	—	—	—	1400	300	4	—
NIGHT TURN	8	184	16.0	—	—	76	68	72	15	73	18	73	68	—	—	70	—	—	100	72	73	69	—	—	155	155	—	—	—	—	1400	300	4	—
	11	202	14.0	—	—	78	70	73	13	74	16	72	68	—	—	72	—	—	102	73	74	68	150	—	—	155	—	—	—	—	1400	300	4	—
	2 A.M.	200	13.0	—	—	76	67	71	12	71	15	68	65	—	—	70	—	—	98	71	72	67	150	—	—	156	—	—	—	—	1400	300	4	—
	5	180	14.0	—	—	72	66	70	13	68	16	67	64	—	—	70	—	—	98	71	72	65	150	—	—	155	—	—	—	—	1400	300	4	—
AVERAGES		199	15.0	—	—	78	69	73	14	73	18	72	69	—	—	71	—	—	100	72	73	69	150	—	155	155	—	—	—	—	1400	300	—	—

SUPPLIES RECEIVED

OBSERVATIONS

Started Bibus pump at 8:00 A.M. shutting down centrifugal pump.
Repacked centrifugal pump and started again, shutting down Bibus pump at 11:00 A.M.
Started Theisen washer No.1 at 3.00 P.M. shutting down washer No.3 on account of
bricklayers working around No.3 motor.

Martin Mullin Day Operator
Mike Smolinski Night Operator

FIG. 24 DAILY REPORT FORM FOR GAS-CLEANING PLANT

60 Suppose, for example, the dust determinations of a certain day, or of several consecutive days, show a much higher amount of impurities in the engine gas than usual. Steps to remedy this condition may be taken immediately. As a rule an increased amount of water in the preliminary washing plant, or on the Theisen washers, will have the desired effect; or an additional washer can be started, thereby decreasing the load on each unit and giving the gas an additional scrubbing. Without the daily record this increase in dust might not be noticed until trouble arose in the engines; or a clogging of the wet scrubbers or of the gas flues might not be noticed until the effect of a restricted gas passage was shown in the reduced output of the power plant.

61 For example, on consulting the daily records in September and October 1908, it was noticed that the gas pressure between the two wet scrubbers was considerably lower than that in the collecting flue, a state of affairs particularly annoying at that time as another period of insufficient gas supply was on hand. It was first thought that the hurdles in wet scrubber No. 1 were clogged by dust bridging over between slats. Simultaneous readings of the pressure gages on either side showed a difference of $1\frac{1}{4}$ in. to $2\frac{1}{8}$ in. After flushing the scrubber for 30 minutes by opening wide all the top sprinklers and side flush-outs, which are situated half way between top and bottom of the scrubber, and using 1,800 gal. of water per minute, this difference did not disappear, indicating beyond a doubt that no obstruction had occurred in the scrubber. It was finally found that the scrubber inlet pipe was nearly filled with mud at the point where it turns at a slight angle into the wet scrubber shell and a heavy stream of water soon removed the obstruction.

62 In March 1909 the amount of flue dust in the dry cleaned gas increased rather suddenly from 0.56 grains per cu. ft. on March 3 to 1.53 grains per cu. ft. on March 5. The amount of water on wet scrubber No. 2 was increased from 400 to 500 gal., and decreased from 400 to 350 gal. per min. on the Theisen washers. The effect was a material improvement in the wet scrubber efficiency, while the amount of dust in the fine gas was hardly affected. This is illustrated in Fig. 25, showing the daily averages of the dust contents in the gas, etc. In this manner the total amount of water for wet scrubbers and Theisen washers as well as the relative quantities for scrubbers No. 1 and No. 2, were changed frequently during the year to conform with the demands indicated by variations in the recorded results. The methods and instruments used to obtain these records are given in Appendix No. 3.

RECORDS AND RESULTS OF OPERATION OF THE DRY-CLEANING PLANT

63 Before the existence of the dry cleaning system at the blast furnaces the two dry dust catchers in the preliminary gas-cleaning plant proved very satisfactory in operation and efficiency, and the effect of unlined gas flues and dust catchers on the reduction in temperature of the gas was greater than had been anticipated. The temperature at which the gas leaves the furnace top averages about 400 deg. fahr. with the furnaces in normal operating condition. This temperature may, however, reach 700 and 800 deg. when abnormal conditions of operation are caused by high coke consumption, irregular working, etc., and furthermore when special grades of iron, such as ferrosilicon or spiegeleisen are produced. When formerly all furnaces discharged their gas directly into the brick-lined overhead flue the temperature of the gas at the entrance of the gas-cleaning plant was considerably higher. Thus in 1908 the average temperature at the main water seal, according to Bristol pyrometer records, was as follows in degrees fahrenheit:

March	April	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg.
650	500	483	531	426	410	303	312	299	329	425

The drop in temperature of the gas, due to radiation of heat through the walls of unlined piping and dry dust catchers, was quite pronounced. See Appendix 4, Table 1. In round numbers about 50 per cent of the sensible heat carried by the gas into the dry-cleaning plant was removed by radiation.

64 An attempt was made to determine the number of B.t.u. radiated per hour per square foot of radiating surface of the dry dust catchers to obtain a basis for future calculations. For five different days the B.t.u. loss per square foot of radiating surface per degree difference in temperature per hour was 1.29; 0.99; 1.105; 1.11; 1.33; with an average for all observations of 1.165.

65 After the dry dust catcher system at the blast furnaces was put in operation conditions changed considerably, as the gas passing through the voluminous unlined dry dust catchers and the overhead gas main, the brick lining of which had been removed in April 1909, lost so much heat by radiation that it entered the gas-cleaning plant at a much lower temperature than before. This temperature is very uniform at present, averaging about 300 deg. fahr. The heat-radiating effect of the dry-cleaning plant, however, is maintained, reducing the average temperature of the gas before it enters the wet-cleaning

plant about 56 per cent as shown in Fig. 26, which gives the average monthly figures for 1909. Since the cooling effect takes place without the use of water, it is obtained entirely without cost.

66 This cooling of the gas to a temperature considerably below 212 deg. in 1909 caused heavy condensation of moisture in the pipes and

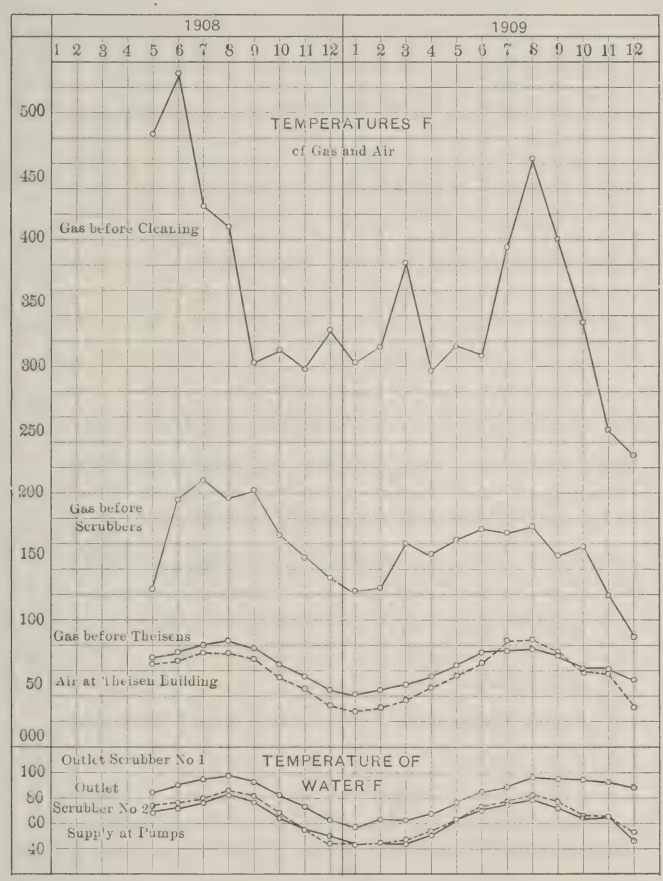


Fig. 26 Temperatures of Gas, Water and Air (Monthly Averages)

dry dust catchers. This proved to be of considerable value in the process of gas cleaning, since the finer particles of dust were weighed down by microscopical globules of water and thus more easily separated from the gas. The dust removed in the dry cleaning plant appeared mostly in the form of mud, which accumulated at all points of

change in direction of the gas flow, and especially in the dry dust catchers where the velocity was very small. The difficulties encountered in trying to remove this mud during the operation of the plant formed one of the reasons for abandoning the dry dust catchers at the gas-cleaning plant. As outlined in Appendix No. 3, several attempts to obtain reliable data concerning the dust-removing efficiency of the dry-cleaning plant proved futile, but an idea can be gathered from the fact that on an average a carload of dry dust weighing about twenty-five tons was removed every other week, while a large quantity was blown away by the wind.

67 The condition in which the gas was delivered to the next stage of cleaning is shown in charts (Figs. 25, 27 and 28) and in Appendix 4, Table 2, wherein the monthly averages as well as the daily variations of the dust contents in the dry cleaned gas are given. These curves and particularly Fig. 28, which gives the daily amount of flue dust in dry cleaned and clean gas for the period from August to December 1909, drawn to a larger scale, indicate quite violent fluctuations which are due to different operating conditions of the blast furnaces.

68 Heavy slipping will naturally increase the dust contents in the raw gas beyond measure. It has an effect very similar to an explosion, as the sudden upheaval of the stock in the furnaces causes a momentary rise in the gas pressure, illustrated on pressure chart Fig. 1. The velocity of travel of the gas through the pipe lines after a slip can easily be observed in the rapidity with which clouds of flue dust belch forth from boiler stacks, nearly 1,000 ft. from the source of disturbance, only a very few seconds after the slipping. Besides the momentary increase in the quantity of dust caused by slipping the furnaces, considerable amounts are added from the deposits of flue dust in pipe lines, dust legs and dry dust catchers, accumulated for hours and days, and disturbed by the sudden high velocity of the gas.

69 The nature of the furnace product has a great deal of influence on the quantity of flue dust produced. While Bessemer and basic furnaces produce about equal amounts in the plant under discussion, ferrosilicon and spiegel furnaces make very much more, which moreover is very fine and cannot easily be removed from the gas—especially not by dry cleaning alone. Thus for instance a sudden increase of 270 per cent in the dust contents in the dry cleaned gas occurred in March 1909, due to the following cause: For several months previous the raw material charged into the furnaces had been considerably "watered" to reduce top temperatures and flue dust losses. On

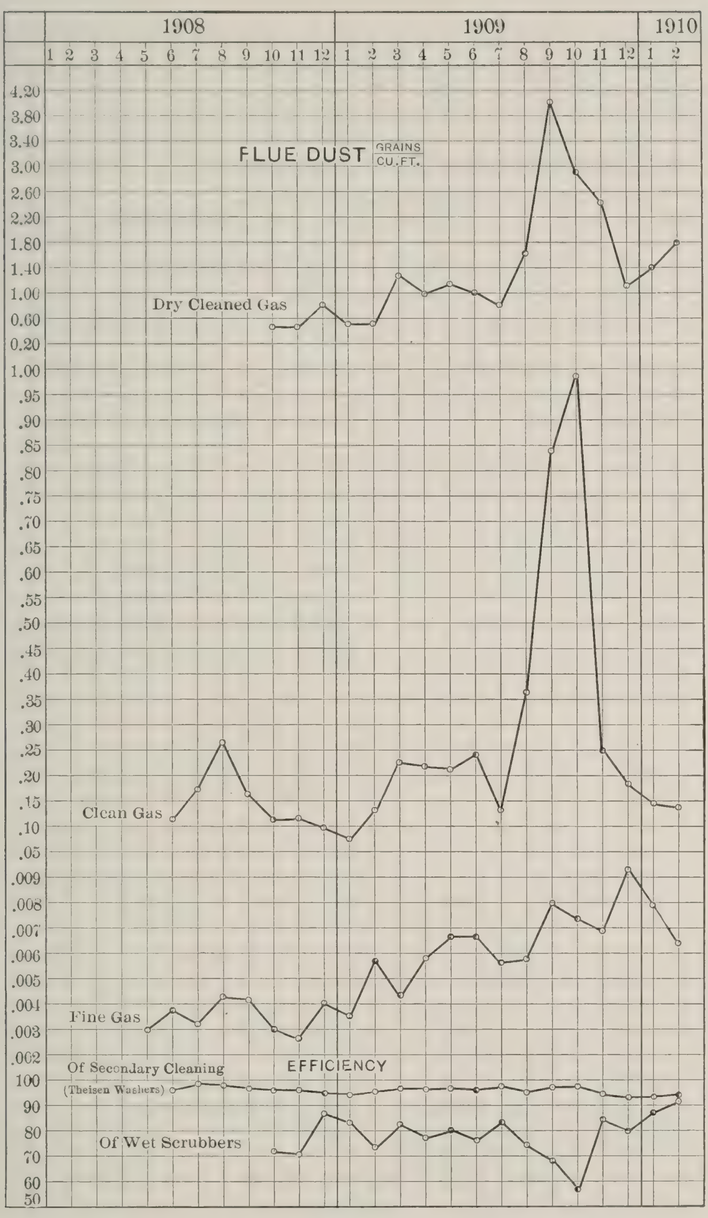


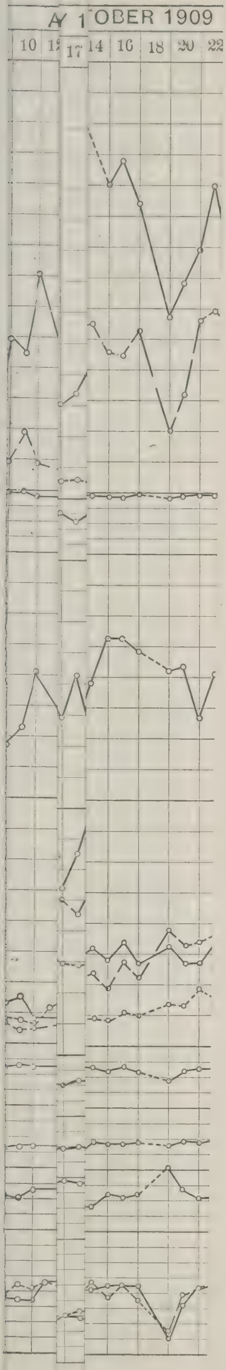
FIG. 27 FLUE DUST IN GAS AND EFFICIENCIES OF CLEANING PLANT (MONTHLY AVERAGES)

February 28, 1909, the watering of the stock was suddenly discontinued. The result was an increase of nearly 100 per cent in the dust contents in the dry cleaned gas, as shown in Fig. 25. It begins with the day ending March 4 and shows great fluctuation, while after March 24 a sudden decrease occurs and the amount of dust for the rest of the month shows considerably more uniformity.

70 The reduction was due to the fact that on March 24 the watering of the stock was resumed in a moderate way. The average amount of flue dust in the dry cleaned gas in February was 0.4787 grains, against 1.2951 grains per cu. ft. in March, the corresponding increase of dust in clean gas being from 0.1224 grains in February to 0.2238 grains per cu. ft. in March, or about 200 per cent. The quantities of gas cleaned per minute were almost exactly the same, namely 14,765 cu. ft. in February and 14,717 cu. ft. in March, so that the records are directly comparable. It cannot be estimated even, how much dust the raw gas contained during March, but when the dry dust catchers were opened for examination on April 3 it was found that the mud in the bottom cones had accumulated to the umbrella, while large dust and mud deposits were discovered in the piping even, the latter having lost its self-cleaning qualities due to the nature of the deposits, which in their muddy condition refused to slide down into the dust legs.

PERFORMANCE OF WET-SCRUBBING PLANT—COOLING AND CONDENSING EFFECT

71 The preliminary wet-scrubbing plant was particularly successful and efficient, and the operation of the wet scrubbers has been continuous ever since starting in November 1907. Several examinations of the wet scrubbers took place, at times when the gas power station was shut down entirely, and it was invariably found that both were in perfect condition. Aside from a thin coating of slimy flue dust, which seems to have penetrated into the pores of the wood, there was no sign of any deposit on or around the hurdles. Since oxygen is practically entirely absent, rotting of the woodwork is impossible. From observations of the condition of the wet scrubbers the conclusions may be drawn that it is unnecessary to be particular in selecting the quality of lumber for manufacturing the slats and that dressing it all over could possibly be dispensed with. The distance between the slats could be made very much smaller than in the two original wet scrubbers, and a spacing of about three inches was adopted for



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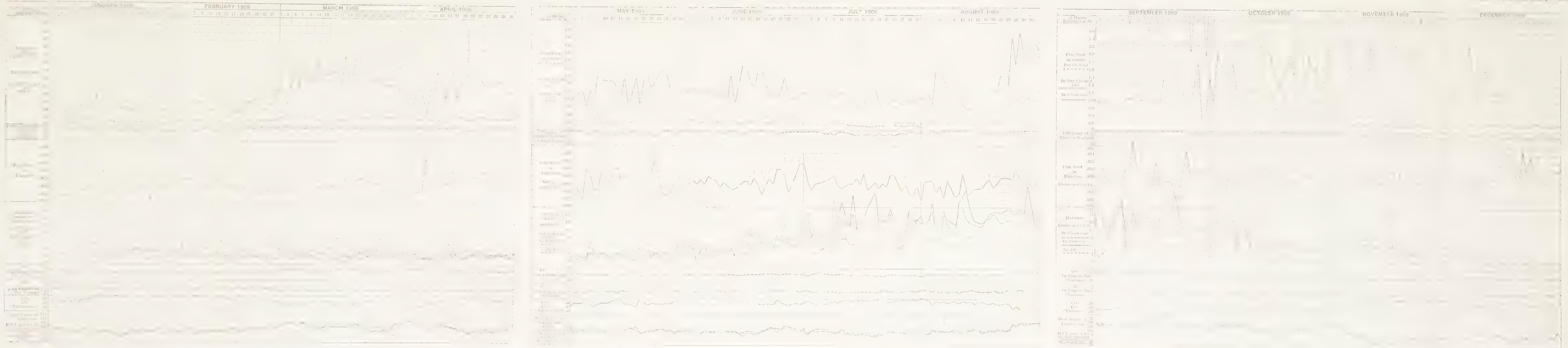
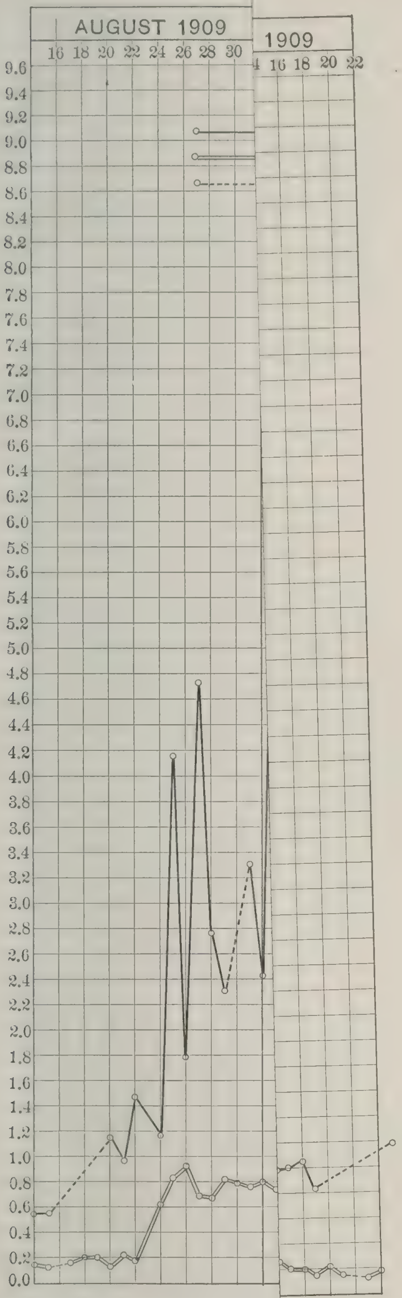


Fig. 25 Condition and Composition of Gas Daily (Averages)


AUGUST 1909
1909
16 18 20 22 24 26 28 30
4 16 18 20 22
9.6
9.4
9.2
9.0
8.8
8.6
8.4
8.2
8.0
7.8
7.6
7.4
7.2
7.0
6.8
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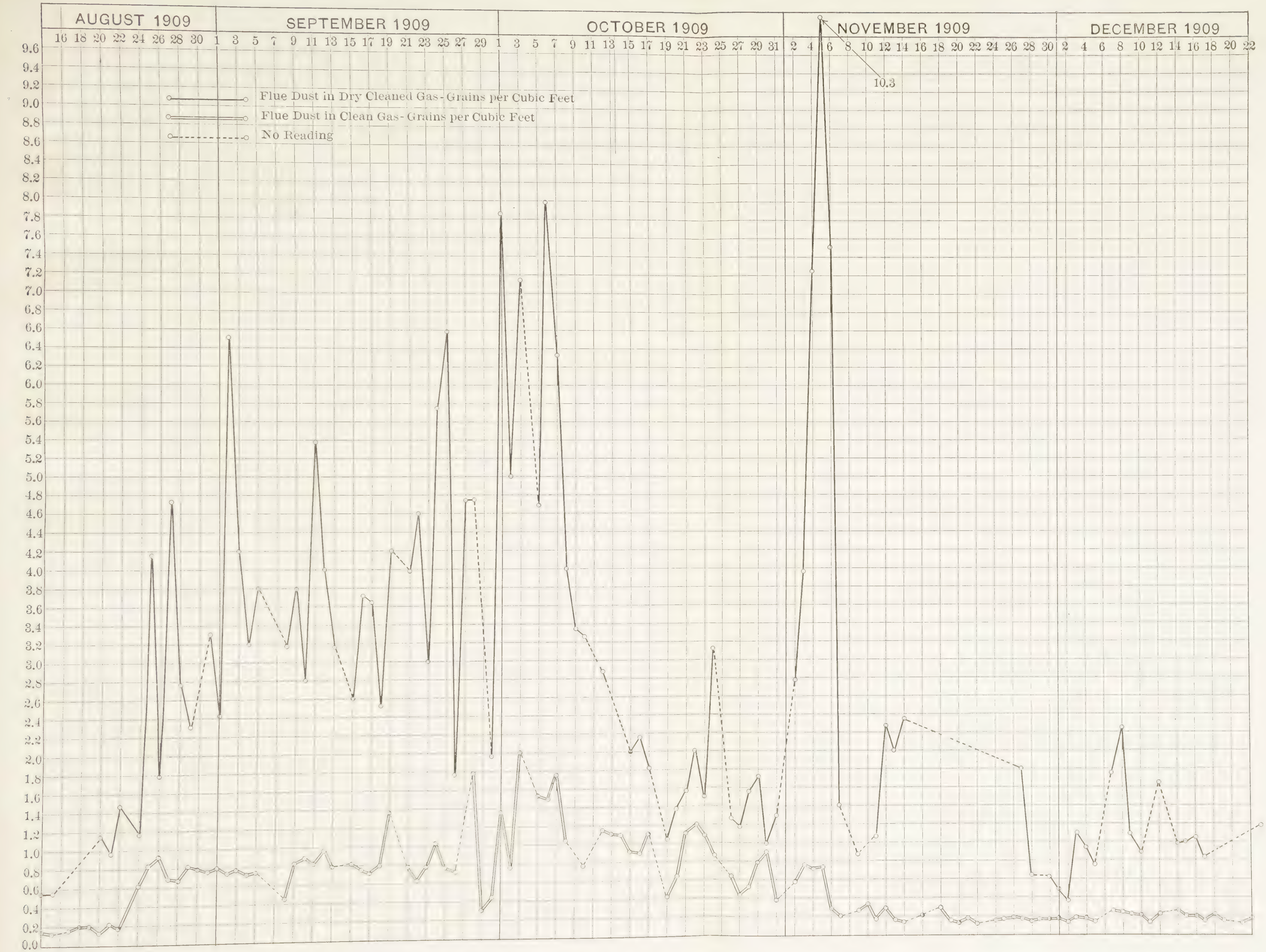

Fig. 28 Flue Dust in Dry Cleaned and Clean Gas

the two new ones. No difficulties of any description were encountered in the operation of these scrubbers, and the overflow arrangement from the water sealed basins, shown in Fig. 11*b*, as well as the disposal of the dirty washing water never caused any trouble.

72 In Fig. 26 and in Appendix 4 (Table 3) are shown the effects of wet cleaning upon the temperature of the gas and the corresponding fresh and waste water temperatures. A comparison of the curves shows that the gas was cooled practically to water temperature, and records show that this cooling action is nearly limited to the first wet scrubber, as the temperature of the gas between the wet scrubbers was only a few degrees higher than the clean gas temperature. The average temperature of the clean gas for the first half year was 54.7 deg., while the temperature of water supply was 53.3 deg. for the same period. For the second half of 1909 these temperatures were 67.4 and 66.7 deg. respectively. The yearly average temperature of the clean gas was 61.1 deg., while the yearly average temperature of the water supply was 60.0 deg. The temperature of the waste water from scrubber No. 1 was on an average 20 deg. higher than the temperature of the fresh water, while the water from wet scrubber No. 2 did not exceed the average water supply temperature more than 1.7 deg. fahr. The first wet scrubber naturally removed the bulk of the dust, as was indicated by the muddy, black appearance of the waste water, but a good share of the cleaning was done by the second scrubber, judging by the reddish-brown color of the water discharged.

73 The cleansing efficiency of the wet scrubbers, that is, the ratio of the amount of dust removed by the scrubbers to the total amount which they receive is given in Figs. 25, 27 and 28 and in Appendix 4, Table 2, showing the amount of flue dust in clean gas, with its daily and monthly variations. The average efficiency of the wet scrubbers was nearly 80 per cent during the first, and 78.8 per cent during the second half of 1909, while the average yearly efficiency reached 79.3 per cent. The drop in the second half, and particularly in September and October, is due to the ferrosilicon and spiegeleisen runs on blast furnace No. 1. If the relatively high amount of dust in the dry-cleaned gas for the same period is considered, as well as the fact that silicious dust is exceedingly difficult to remove, this reduced efficiency is not surprising.

74 Of greatest interest is the effect of wet cleaning on the amount of moisture in the gas. This information is given in Fig. 25 and Fig. 29. While the quantity of water used at the wet scrubbers was considerable, averaging 82.8 gal. per 1000 cu. ft. of gas cleaned, the aver-

age amount of moisture in clean gas was only 6.62 grains per cu. ft. with a maximum of 13.243 grains in August and a minimum of 2.61 grains in April. By comparing these figures with the corresponding average amounts of moisture in atmospheric air, an interesting coincidence will be noted, as the maxima in both cases occurred in August while the minima obtained in April. In Appendix 4 are detailed results of tests made to determine the cooling and condensing effects of the wet scrubbers.

PERFORMANCE OF SECONDARY CLEANING PLANT

75 The Theisen gas washer installation in the secondary cleaning plant was very successful, both in regard to the mechanical operation and the cleaning efficiency of the washers. Theisen washer No. 1, started in 1907, was opened for examination on February 6, 1909,

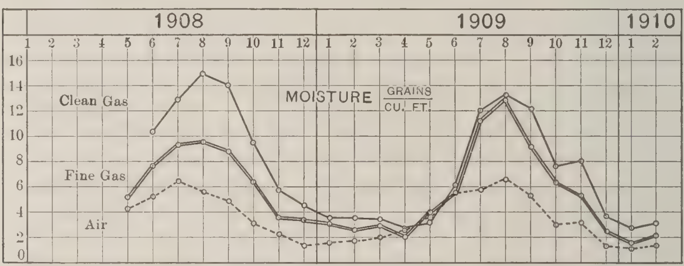

FIG. 29 MOISTURE IN GAS AND AIR (MONTHLY AVERAGES)

after about 7,400 hours of operation. The condition of this washer was as follows: The drum was almost perfectly clean, the longitudinal vanes showing a coating of soft mud about ½ in. in thickness and 10 in. in length on the back side near the gas inlet. The total amount of mud and dust on the vanes when dried filled a 3½-gal. water pail twice, and, except to repaint the drum, the washer needed no attention. The paint was worn off the front of the vanes only, especially on the outer edges. The wear of the water on the longitudinal vanes, due to its velocity and its contents of granulated cinders, was quite noticeable at the points were the water happened to impinge. On account of a slight construction defect and the inadvertent use of hard high-carbon steel in their manufacture some of these vanes cracked along their rivet holes and needed replacing. Softer low-carbon steel has since been used and the longitudinal vanes braced, as shown in

Fig. 19, and this trouble has not again occurred. The wear in the babbitt-lined bearings allowed the shaft to lower about 1/32 in.

76 The wire netting was absolutely clean, while somewhat corroded in places on the bottom, and the claim that this washer is self-cleaning was fully substantiated by the examination. Theisen washer No. 2 was opened in March 1910 for its second examination after nearly 9,300 operating hours, and its condition was found to be equally satisfactory. The accumulations of mud were very slight as shown in

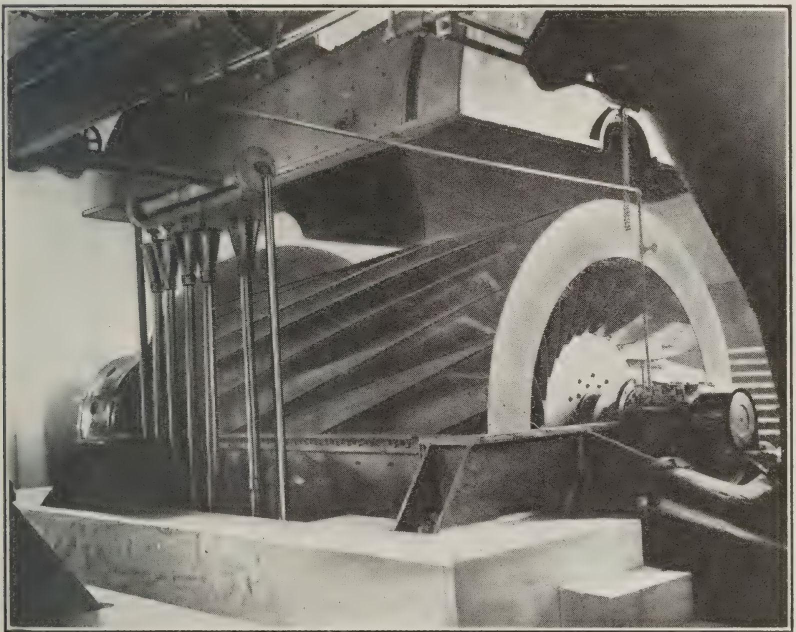

FIG. 30 THEISEN WASHER DURING EXAMINATION

photographs, Fig. 30 and Fig. 31, which were taken immediately after removing the top half of the casing. The washers can run continuously for months without being shut down, except for occasional cleaning of the motors. All smaller repairs on these washers, as well as on the whole gas-cleaning plant, are made by the operators, and the expenditure for lubricants and other supplies is amazingy small.

77 The chart in Fig. 2 shows that the average gas pressure after the Theisen washers was only slightly in excess of the aver-

age raw gas pressure, the difference being about 3 in. of water column. The advantage of this slight pressure difference is obvious, in contrast with the considerably higher pressure given by so-called hydraulic fans. Superfluous pressure for the transmission of the gas to the point of consumption, must be paid for in excess power. The gas temperature at the Theisen washer inlet was practically water temperature, while the temperature of the gas delivered by the Theisen washers was found to be on an average 2 deg. to 3 deg. higher, a difference

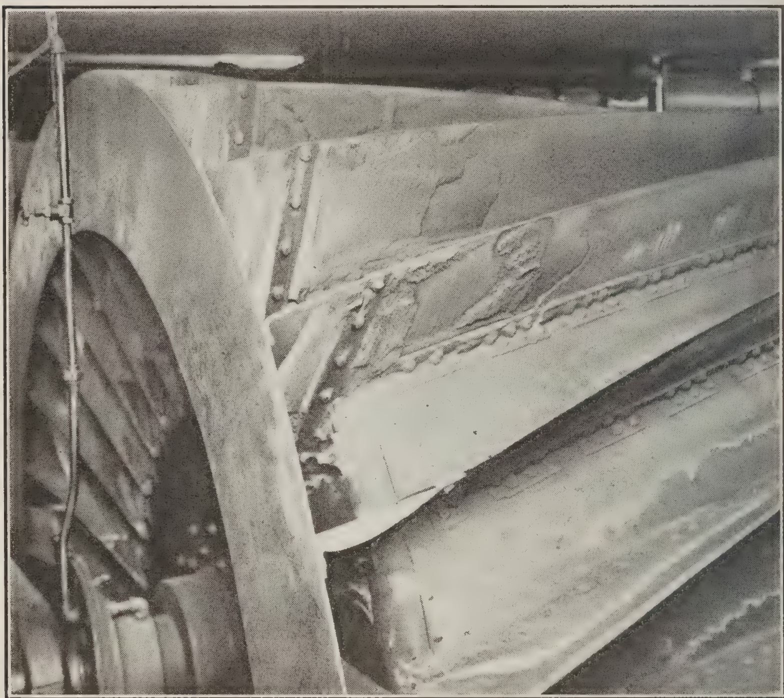

FIG. 31 THEISEN WASHER—DETAIL OF DUST DEPOSIT

explained by the fact that the mechanical work consumed by the Theisen washers must be transformed into heat, which in portion is imparted to the gas. On its way to and in the gas holder this heat is again radiated, so that the gas temperature at the engines practically equals the temperature of water supply, and only a little difference can be observed between the atmospheric temperature and the temperature of the fine gas (See Fig. 26).

78 The performance of the Theisen washer as a gas cleaner, considering the variety of conditions of furnace operation, was beyond reproach. By referring to the daily and monthly averages, given in Table 4 of Appendix 4 and plotted in the charts in Fig. 25 and Fig. 27, it will be noted that the efficiency of the secondary cleaning plant, that is, the ratio of the amount of dust removed by refining to the total amount contained in the clean gas leaving the preliminary washing plant, averaged 98.1 per cent. While this figure in itself is remarkable, it is of much more importance that this efficiency, as shown by the close coincidence of the monthly average figures, was exceedingly uniform, varying from a maximum of 99.1 per cent in October, to a minimum of 95 per cent in December 1909. The average efficiency for the first half-year was 97 per cent, while the corresponding figure for the second half reached 98.7 per cent. The great variations to which the amount of flue dust in dry cleaned gas was subjected during the year, especially however in March and April and during the period from the latter part of August until the middle of December, did not particularly affect the efficiency of the secondary cleaning plant, since in September and October, when the amount of flue dust in dry cleaned gas averaged 3 to 4 grains per cu. ft., the efficiency of the secondary washing plant shows a very marked uniformity—the average from August 25 until November 5 being 98.87 per cent—while the efficiency of the wet scrubbing plant gradually decreased from 83 per cent in July to 75 per cent in August, 69 per cent in September, and 57 per cent in October. The amount of flue dust in fine gas during these months was, of course, higher than in any previous period, but the efficiency of refining was nearly a constant maximum, irrespective of the dust conditions of raw and clean gas and of the quality of the flue dust.

79 During September and October, when furnace No. 1 made its ferrosilicon run, the color of the waste water from wet scrubbers and Theisen washers was very milky, and the dust so high in quantity, and so peculiar in quality, that considerable accumulations occurred in the clean gas main and at the inlet gate valves of the Theisen washer. The silicious flue dust seemed to set like cement, and lumps of considerable size and of great hardness had to be removed with a bar, from the inlet gate valves of the Theisen washer. Nevertheless the efficiency of the secondary washing plant during these two months was higher than during any other period.

80 In the previous paragraph the term "efficiency of the secondary cleaning plant" was used deliberately in order to indicate that in

this remarkable showing gas mains and gas holder participated. The clean and fine gas flues, connecting wet scrubbers, Theisen washers and power station gas holder, have a combined length of approximately 1000 ft., and a diameter of 5 ft. 6 in. and 5 ft. respectively. The total quantity of gas passing these mains per minute averaged 16,900 cu. ft., with a maximum of 20,300 cu. ft. in October, and a minimum of 13,600 cu. ft. in January. The velocity of the gas while traveling through these pipes did not exceed therefore 14 ft. per sec. in the clean gas main, and 17 ft. per sec. in the fine gas main. At this low velocity some impurities will undoubtedly drop out in the clean gas main, with the finely divided water which is carried along by the gas. The fine gas main, however, was always found practically clean; and while an examination of the gas holder tank has not been made, it is not believed that any great quantities of flue dust would be found in the bottom of the tank, as the dust in the fine gas is so impalpable that the fine gas burns with absolutely clear blue flame. However, in order not to credit the Theisen washers alone with removing 98 per cent of the dust from the clean gas, the efficiency is stated as embracing the secondary washing plant as a whole, or the combination of gas pipes, Theisen washers, water separators and gas holder.

81 That the Theisen washers must be credited with the bulk of the work is proven by the test summarized in Table 5 of Appendix 4. From the results given it will be seen that the efficiency of the clean gas main averaged 16.5 per cent while the fine gas main and the gas holder removed only 6 per cent of the dust delivered by the Theisen washers, or 0.23 per cent of the dust in the clean gas, proving that but little cleaning is done by gravity and reduced velocity. The Theisen washers had nearly 95.5 per cent absolute efficiency during the week when the tests were made, while their relative efficiency, based on the amount of dust in clean gas, was 79.64 per cent. It is safe to assume that similar conditions prevailed during the year 1909 and that the same relative proportions hold true at any time. The inconsistency in some of the results obtained after the Theisen washers and after the gas holder, which would indicate the impossible condition that the gas picked up a certain amount of dust on its way from washers to holder, is due to the small quantities on the shell of the Brady filter, which have to be dealt with in fine gas and cause errors in observations.

82 Since starting the gas power plant in 1907 the absolute amount of dust in the fine gas naturally continued to increase gradually, corresponding to a similar increase in the amount of gas cleaned per

minute, shown in Fig. 32. At the beginning of operations the gas engines received gas of a degree of cleanliness excessive for practical purposes. Thus the average dust contents in the fine gas in the second half of 1908 was only 0.0036 grains per cu. ft., or 0.0077 grams per cubic meter, which is much less than the average usually guaranteed. An excessive purification of blast furnace gas, even for engine purposes, is unwarranted, because the atmosphere in a steel plant is usually very dirty, and it seems quite out of place to purify the gas to a higher degree of cleanliness than the combustion air, unless the latter is to be subjected to a similar cleaning process. Tests made in July 1909 to determine the quantity of impurities contained in the air near the air intake of the gas engines showed the amount to be between 0.0005 and 0.0052 grains per cu. ft. with an average of determination (given in

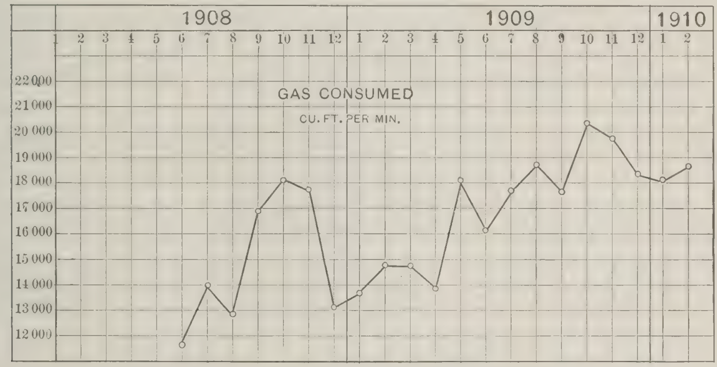

FIG. 32 GAS CONSUMED, CU. FT. PER MIN. (MONTHLY AVERAGES)

Table 6, Appendix 4) of 0.00346 grains per cu. ft. The amount of dirt in the air is therefore by no means a negligible quantity. The detrimental character of these impurities is also apt to be under-estimated. In Appendix 4, Table 7 is a comparison of the chemical analyses of samples of the deposit on the gas and air dampers in one of the gas engines, taken in February 1909. It is to be noted that the gas never comes in contact with the air dampers, so that the samples fairly represent the accumulation of dirt deposited by the combustion air.

83 A comparison of the four analyses shows the surprising fact that the amount of iron in the dust sample taken from the air dampers is about twice as great as in the dust deposited on the gas dampers. If this is true—and there is no room for doubt, as the samples represent the accumulations of more than one year, the dampers never having

been cleaned—it follows that appreciable quantities of iron, sand and coke enter the engines with the combustion air.

QUALITY OF FLUE DUST

84 The chemical composition of the flue dust removed from the gas at various stages of the cleaning process was made the subject of analysis in March 1908. Samples were taken at the following points: dry deposit from the main water seal and collecting flue after dry dust catchers, suspended matter secured by evaporation of samples of the waste water from wet scrubbers No. 1 and No. 2 and the Theisen washers. The results of this test are shown in Table 8, Appendix 4, all analyses giving metallic iron and manganese as Fe and Mn, while the constituents exist in the form of oxides. Fixed carbon is mostly coke, and the volatile is the CO_2 from limestone. This table shows that the relative amounts of silica, alumina, lime, etc., increase gradually, more and more of the heavy impurities such as Fe dropping out the further the cleaning process progresses.

85 A similar test was made in March 1909, with the difference, however, that an attempt was made to determine the quality of the dust remaining in the gas while passing the various stages of gas cleaning. In this case the method of securing samples consisted in connecting Brady filters at the various points and turning the gas on simultaneously at all filters. Irrespective of the size of the gas sample, the cartridges were not removed until sufficient quantities of dust had collected for quantitative and qualitative analyses. It was impossible to obtain a sufficiently large sample for analysis, on the Brady filter installed after the Theisen washers, for while the filter passed the gas very rapidly for a few hours, the flow gradually diminished as the dust in the fine gas closed up the pores of the paper. After a 36-hour run the flow through the filter had stopped completely and the dust collected on the Soxhlet tube appeared as a dull gray coating, which could not be removed. The results of this experiment are given in Table 9, Appendix 4.

86 The relative amount of SiO_2 carried in the gas is practically constant before any wet washing takes place, but the percentage increases in the clean gas, with decreasing relative contents of iron. The wet scrubbers remove the bulk of the iron dust, while the lighter impurities are carried over into the Theisen washers. A sample of the dust deposited in the gas pipe on one of the gas engines immedi-

ately before the gas enters the cylinders at the end of its travel, taken March 15, 1910, was analyzed as follows:

SiO_2	Al_2O_3	Fe	CaO	MgO	Mn	Vol.
36.20	7.53	7.18	12.50	0.90	0.49	34.32

The high percentage of silica, lime and volatile matter, and the very low contents in iron, are noteworthy, and show that silicious dust, lime and coke are carried by the gas much farther than iron, which is removed to the greatest extent in the first stages of cleaning.

87 Due to sulphur in the coke, blast furnace gas contains a certain amount of this impurity in the form of H_2S and SO_2. Its pres-

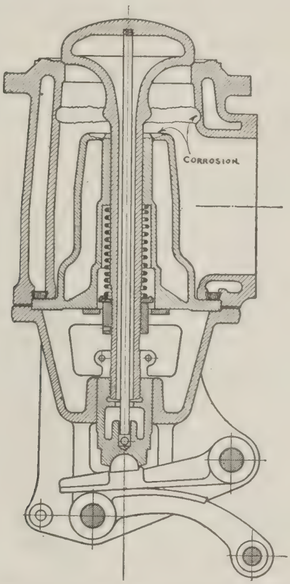

FIG. 33 CORROSION OF VALVE CASING BY SULPHUR IN GAS

ence was discovered after some length of operation of the gas engines. It was found that several of the exhaust valve casings showed very pronounced corrosion, at a place indicated in Fig. 33 and it was concluded that some chemical action had taken place. Two tests made Feb. 2, 1909, showed the presence of 0.0416 and 0.0407 grains per cu. ft. of sulphur in the gas. It was further discovered that the fine flue dust accumulating in thin layers in the gas passages to the engine cylinders, can be lighted and glows with a blue color, unmistakably giving out SO_2 vapors. The corrosion in the exhaust valve chambers was caused by water leaks from the original inner exhaust valve guides,

which cracked as indicated in the illustration. The water coming in contact with the hot exhaust gases formed H_2SO_4, which impinging on the wall of the exhaust valve chamber, had the corrosive effect. The presence of sulphur in blast furnace gas for this reason prohibits the use of sheet-iron exhaust pipes and mufflers, while corrosion has never been observed when cast iron is used. It was attempted, some time ago, to utilize the waste heat from the exhaust of these engines to raise low-pressure steam for use in the heating system of the power house, in wrought-iron pipe coils arranged inside the cast iron mufflers, and while the results were very satisfactory from the standpoint of heat transmission, this heating system was a failure on account of the rapid corrosion to which the heating coils in the mufflers were subjected.

88 The amount of moisture remaining in clean gas, fine gas and air, as well as the variation from day to day and from month to month, are shown in Figs. 25 and 29.

89 While the Theisen gas washers receive the gas at practically water temperature, so that a condensation of water vapors by cooling is improbable, the amount of moisture in the engine gas is nevertheless lower than in the clean gas, and this in spite of the exceedingly intimate contact between gas and washing water in the Theisen washers. The indications are therefore that the Theisen washers remove not only dust, but moisture as well, probably by the action of centrifugal force, which throws gas and water vapors against the circulating water film on the inside of the stationary casing, thereby drying the gas mechanically. Furthermore a great deal of moisture is being deposited in the fine gas main and gas holder. The average moisture in the gas delivered to the engines was 3.39 grains for the first half, and 7.85 grains for the second half of 1909, with a yearly average of 5.62 grains per cu. ft. Comparing these figures with the corresponding values in clean gas and atmospheric air, it will be seen that the average moisture in the engine gas for the year is 60 per cent higher than the average moisture in the atmosphere, which is considered very favorable in view of the high moisture contents in raw gas and the large quantity of water which is brought into such intimate contact with the gas.

90 The effect of the sun beating down on gas mains and gas holder is very noticeable in the summer months, accounting for the high moisture contents in July, August and September. Pipes and holder become quite hot and the finely divided mist in the gas is rapidly evaporated, as indicated by a drip installed near the venturi meter.

Drops or small streams of water are freely discharged in the winter months, while the dripping ceases as soon as the atmospheric temperature exceeds about 78 deg. fahr. The moisture carried with the gas into the engine cylinders has not given cause for trouble at any time.

WATER AND POWER CONSUMPTION OF CLEANING PLANT

91 Fig. 34 shows the average monthly consumption of water since July 1908 in gallons per thousand cubic feet of gas cleaned, as

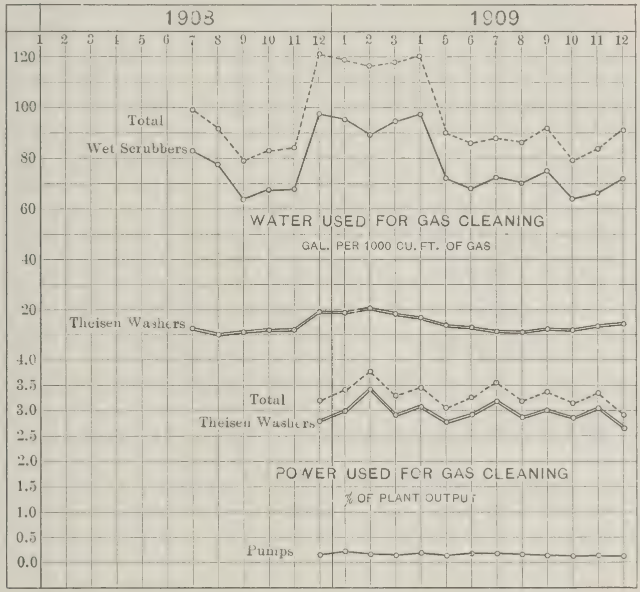

FIG. 34 WATER AND POWER CONSUMPTION OF GAS CLEANING PLANT

established from daily weir measurements. It is to be noted that the water consumption of the wet scrubbers is about four times as high as that of the Theisen washers, for the same quantity of gas cleaned. The latter varied from 26.1 gal. in February to 16.0 gal. in July, with an average of 21.8 gal. for the first and 17.0 gal. for the second half, and 19.4 gal. per 1000 cu. ft. of gas for the whole year. The corresponding figures for the wet scrubbers were a maximum of 103.2 gal. in April, and a minimum of 68.6 gal. in October, with an average of 91.0 gal. for the first half, 74.6 gal. for the second half, and 82.8 gal. per 1000 cu. ft. for the year 1909.

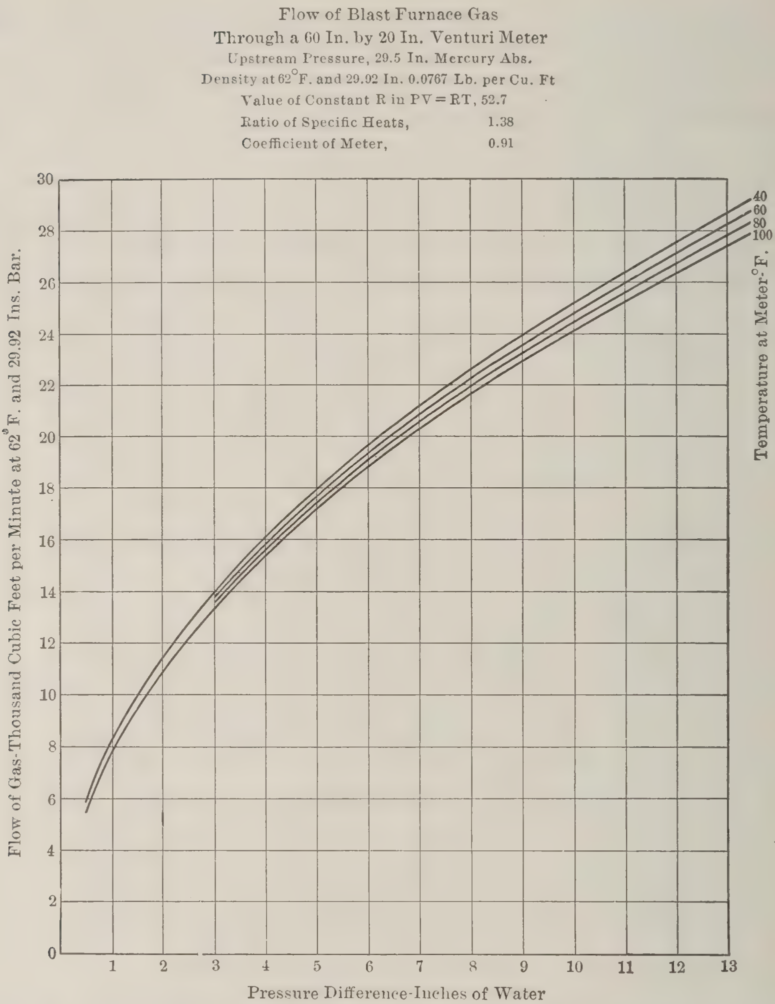

FIG. 35 VENTURI METER CURVES SHOWING FLOW OF GAS FOR DIFFERENT METER READINGS

92 The high water consumption in the first part of the year is due to the smaller quantity of gas cleaned. About the same absolute quantity of water was maintained on the scrubbers. Generally speaking, the quantity of water used in the wet scrubbers could probably be reduced without impairing their efficiency, but it is preferred to use water in excess rather than to run the risk of clogging the sewers, especially since the question of economizing washing water is of no particular importance in a plant located on the lake front.

93 In regard to the amount of power required by the gas cleaning plant, Fig. 34 shows the monthly average power consumption of wet scrubber pumps and Theisen washers, expressed in per cent of the output produced by the gas engines. About 90 per cent of the total power is being used by the Theisen washers, only 10 per cent being

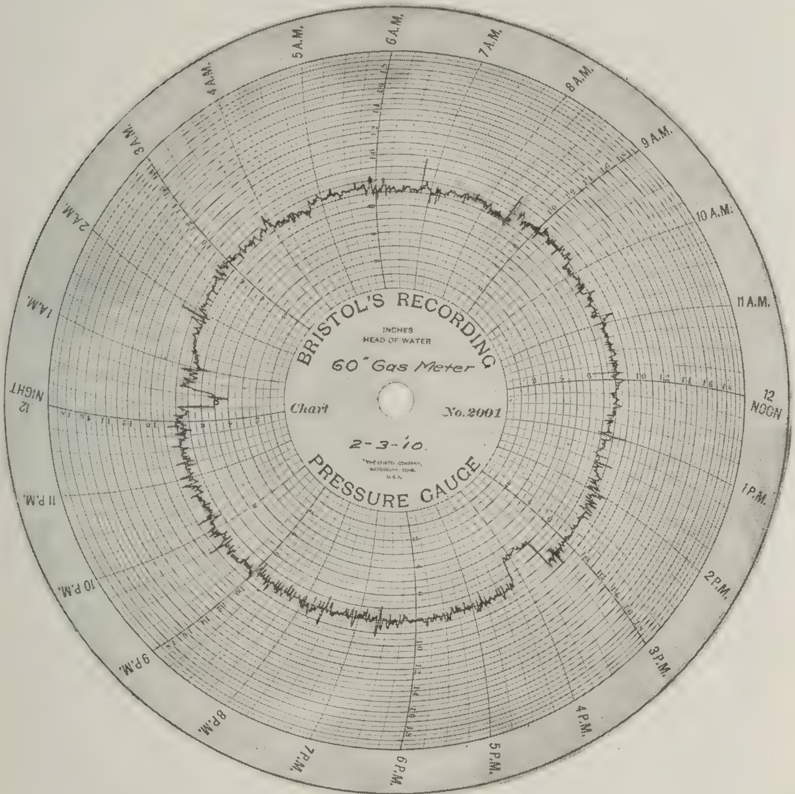

FIG. 36 BRISTOL CHART OF PRESSURE DIFFERENCES, 60-IN. VENTURI METER

necessary to operate the wet scrubber pumps. The average power consumption of the Theisen washers was 2.977 per cent of the total output of the station, and the respective values for the first and second halves of 1909 were 3.00 per cent and 2.931 per cent, with a maximum of 3.44 per cent in February and a minimum of 2.649 per cent in December.

94 These figures are somewhat higher than are often claimed for similar washers abroad, but an average power consumption for the gas-cleaning plant, of from 3 per cent to 3.5 per cent of the total power output of the gas engines, should not be considered excessive in view

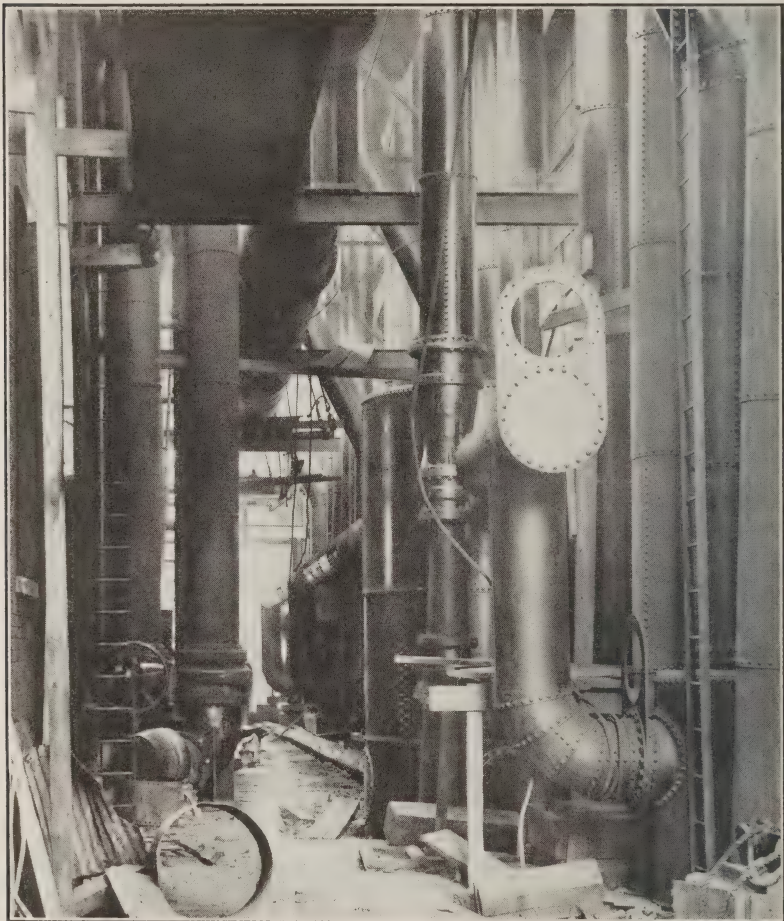

Fig. 37 Exterior View of Test Piping

of the benefit which is being derived from this expenditure. It is worthy of note also that the engine builders who furnished the gas engines for the plant under discussion have never found fault with the physical condition of the gas, while more or less vigorous com-

plaints about "excessive dirt" in the gas are generally advanced as soon as trouble develops in the engines. The necessity of auxiliary machinery is furthermore not characteristic of gas engine installations alone, as boiler-feed pumps, hot-well pumps, dry-air pumps, forced-draft ventilators, mechanical stokers, etc., are indispensable for steam engine plants consuming probably as large a percentage of the power developed.

95 It has been shown in the previous pages that the blast furnace gas was delivered to the gas engines in a highly satisfactory physical condition. The gratifying results of the thorough cleaning and refining of the gas were, that no difficulties which might have been caused by insufficiently cleaned gas were ever experienced in the operation of the gas engines, and these engines never had to be stopped for the specific purpose of cleaning internally the gas valves and gas passages. The amount of dust deposited on internal engine parts was invariably so small that it could be brushed off with the finger, and these engines could undoubtedly operate at full-load capacity a whole year and longer without cleaning the gas inlet passages and cylinders.

96 Blast furnace gas delivered to the power house is charged to operation of the engines at a value based on the price of coal with the cost of cleaning and refining added to the value of the raw gas, which is established on the basis of equivalent heat values. In order to determine the charge made for purified blast furnace gas delivered to the gas power plant, a continuous record is being kept of the quantity of gas blowing to the gas holder, venturi meters being used as measuring instruments. Details of the methods are given in Appendix No. 5. Charts relating to meter measurements are shown in Figs. 35 and 36 of the paper.

THERMAL EFFICIENCY AND OUTPUT OF GAS ENGINES

97 It seemed desirable in connection with the installation of six gas blowing-engines, furnished by three different manufacturers, to provide means for determining the thermal efficiency of each type of engines without interfering with the regular operation of the others. To this end a special venturi meter was installed and connected to a "test" pipe, shown in Figs. 23*a* and 37. By opening the disc valve located inside the overhead gas receiver, which controls the flow of gas through the venturi meter into the large reservoir for equalizing pulsations caused by the intermittent suction strokes, and test piping, and by turning one small spectacle valve while simultaneously

shutting off the direct gas supply from the overhead receiver by filling the water seal, any engine can be operated on measured gas while the others receive gas directly through the branch pipes. Such tests can be commenced at any convenient time, and prolonged and repeated at will. A Bristol differential pressure recorder installed in the blowing engine-room gives continuous records of each test, so that reliable averages can be obtained of the gas consumption under different operating conditions.

98 Fig. 38 shows the monthly averages of the kilowatt output of the power station, of the heat consumption in B.t.u. per kw-hr. and per b.h.p.-hr. based on a generator efficiency of 96.2 per cent at full load, and of the thermal efficiency. The diagram also shows the total operating time of each engine since starting.

99 The thermal efficiency of the plant was very uniform from the middle of 1908 until about May 1909, averaging 23.22 per cent. The drop which began in May and reached a low value in October, was due to certain troubles encountered with gas cylinders and piston rings, etc. These could not be remedied at that time, as on account of the ever-increasing demand of electric power and the lack of a spare unit, it was impossible to shut the gas engines down sufficiently long for a thorough overhauling and for necessary repairs. Doubtless with an additional spare engine the load factor would have been lower than the average for 1909, which reached 72 per cent, but the thermal efficiency would have remained constant—or nearly so—since the necessary repairs, adjustments and changes could have been made on these engines in time, without reduction in the total kilowatt output of the power plant. In spite of this reduced efficiency in the second half of 1909, the average figure obtained for the whole year,—not as the result of one or of several individual tests, but as the fair average of daily observations, proper corrections having been made for the inexact readings caused by dust in the venturi meter in September and October—was 20.8 per cent, with a maximum monthly average of 23.77 per cent in March, and a minimum of 17.8 per cent in October. The highest daily average efficiency in 1909 was 25.7 per cent on March 11.

100 During the year 1909, the engines, while in operation, ran at nearly full-load capacity, the average for the four engines ranging from 93.60 to 99.63 per cent.

101 The values of the total kilowatt output of the gas power plant for each month since regular operation was begun, are plotted in Fig. 39, showing that the maximum output for any month occurred in

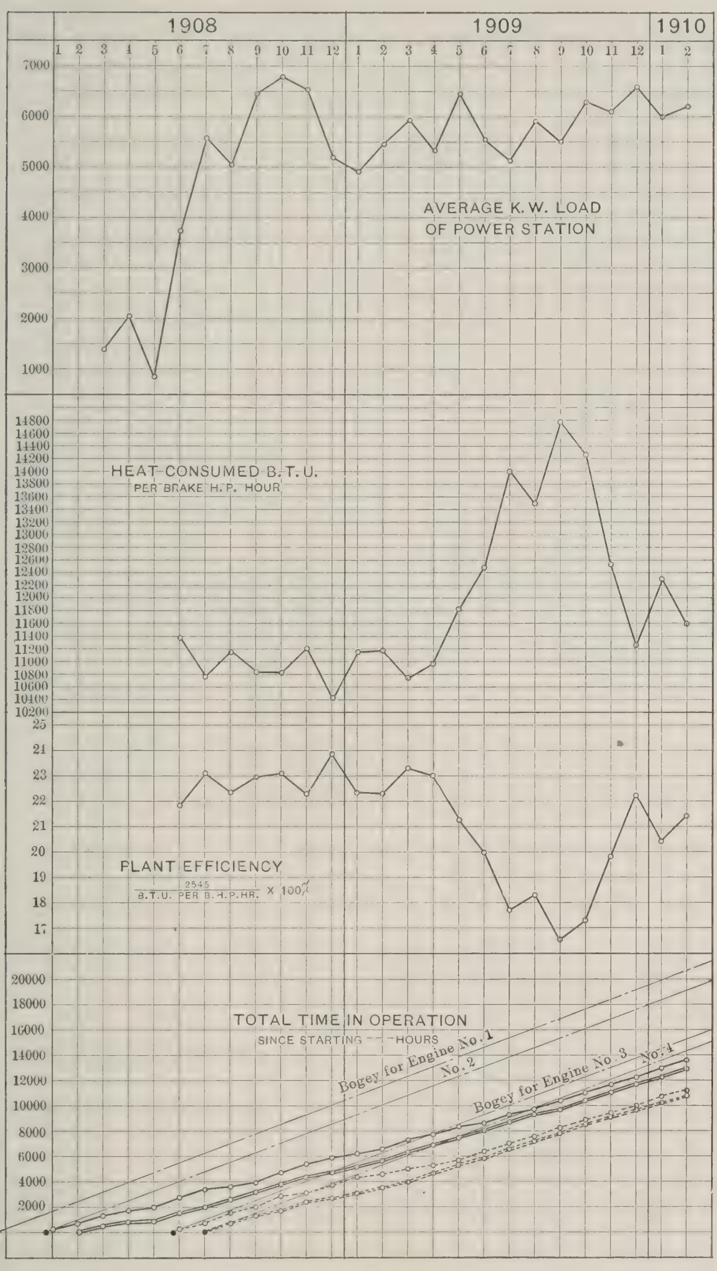

Fig. 38 Average Load, Heat Consumption, Plant Efficiency and Operating Time (Monthly Averages)

October 1908, when 5,000,000 kw-hr. was produced. The output of the gas power plant for the year 1909 was 50,494,100 kw-hr. against 43,953,640 kw-hr. produced by the steam-driven generators. The load factor for 1909 of the gas power plant was 72 per cent, against

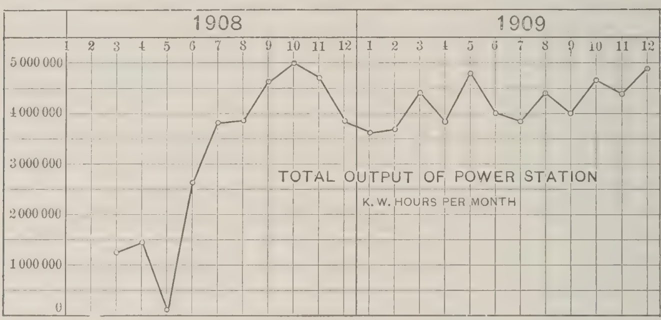

Fig. 39 Total Output of Gas Power Plants (Monthly Averages)

47 per cent for the steam power stations, which have a rated capacity of 10,900 kw. The total output of electric power generated in 1909 was 94,447,740 kw-hr., 53.5 per cent of which was produced by the gas power plant.

APPENDIX NO. 1

MONTHLY RECORDS OF THE POWER PLANT (8000 KW.)

TABLE 1 MONTHLY RECORD OF KILOWATTS PRODUCED PER HOUR

Month 1909	Kw. per Hr.	Per Cent of Capacity	No. of Furnaces in Blast
January	4920	61.5	3
February	5450	68.0	3
March	5940	74.0	2
April	5330	66.5	2
May	6440	80.5	3
June	5550	69.0	4
Average for first half	5600	70.0	
July	5140	64.0	5
August	5930	74.0	5
September	5500	68.5	6
October	6270	78.0	6
November	6080	76.0	5
December	6600	82.5	5
Average for second half	5920	74.0	
Average for 1909	5760	72.0	

TABLE 2 MONTHLY RECORD OF OPERATING TIME

1909	Plant in Operation		Plant down			
			Due to engines		Due to outside causes	
Month	Hrs.	Per Cent	Hrs.	Per Cent	Hrs.	Per Cent
January	426	57	213	28.5	105	14.5
February	456	68	85	12.5	131	19.5
March	529	71	97	13.0	118	16.0
April	467	65	170	23.5	83	11.5
May	609	82	73	9.5	62	8.5
June	556	77	97	13.5	67	9.5
Average for first half	507	70	123	17.0	94	13.0
July	607	82	103	14.0	36	4.0
August	594	80	131	17.5	19	2.5
September	595	83	107	14.5	18	2.5
October	668	90	52	7.0	24	3.0
November	632	88	65	9.0	23	3.0
December	609	82	52	7.0	83	11.0
Average for second half	617	84	85	11.4	34	4.6
Average for 1909	562	77	104	14.2	64	8.8

TABLE 3 MONTHLY RECORD OF LOSSES DUE TO OUTSIDE CAUSES

Month 1909	Plant down			
	Due to operation		Due to lack of gas	
	Hrs.	Per Cent	Hrs.	Per Cent
January	6	5.5	99	94.5
February	43	33.0	88	67.0
March	36	30.0	82	70.0
April	17	20.0	66	80.0
May	61	98.5	1	1.5
June	59	88.0	8	12.0
Average for first half	37	39.5	57	60.5
July	36	100.0	..	....
August	19	100.0	..	....
September	15	83.5	3	16.5
October	24	100.0	..	....
November	23	100.0	..	....
December	78	94.0	5	6.0
Average for second half	33	97.0	1	3.0
Average for 1909	35	55.0	29	45.0

APPENDIX NO. 2

DATA UPON GAS PRODUCED IN THE BLAST FURNACES

Methods of Calculation of Quantity of Gas Produced

The "nitrogen method" assumes that the nitrogen in the air, amounting to 79.3 per cent by volume, passes through the blast furnace unchanged, so that the same quantity must be found in the exit gas of the furnaces. Since the average gas composition is known, the percentage of nitrogen can be determined by difference. The amount of air blown is based upon the revolutions of the blowing engines corrected for volumetric efficiency of the blowing tubs and for 5 per cent assumed loss of air between tubs and tuyeres. Thus, for instance, in August 1909 blast furnace No. 6 produced on natural blast a daily average of 496.5 tons of Bessemer iron, using 37.5 per cent Pocahontas and 26.5 per cent Connellsville coke with an average coke consumption of 2148 lb. per ton of iron. The average gas analysis for the same month was as follows:

CO_2	CO	H	CH_4	N (by difference)
14.23	25.28	4.65	0.23	55.61

B.t.u. per cu. ft. at 62 deg. fahr. = 96.8
B.t.u. per cu. ft. including sensible heat at 500 deg. = 105.3
Temperature of air at blowing engines = 94 deg. fahr.
Cu. ft. of air blown per minute = 40,990
Cu. ft. of air blown per minute at 62 deg. fahr. = 38,610
Average blast pressure = 15.1 lb.
Cu. ft. of air to furnace per minute including 5 per cent loss at 62 deg. fahr. = 36,690
Volume of gas per volume of air at 62 deg. N unchanged = 1,426
Cu. ft. of gas per minute N method = 52,300

According to this calculation No. 6 furnace produced about 153,000 cu. ft. of gas per ton of pig iron made in 24 hours.

2 The nitrogen method, however, is subject to serious errors from the presence of moisture and foreign gases in the air as blown, and from air leakage and inefficiency of tubs. The "carbon method," which is considered more reliable, assumes that the carbon entering the furnace in the form of coke, must reappear in the form of gas, the amount of carbon in the limestone being equal approximately to that in the iron and slag. If A = per cent of carbon in the coke as charged, B = pounds of carbon in 1 cu. ft. of blast furnace gas, the amount of gas produced in cubic feet per minute $= \frac{\text{tons of iron per day}}{1440} \times \frac{\text{coke rate}}{B} \times A.$
Calculating the amount of gas according to this method, furnace No. 6 liberated 51,310 cu. ft. of gas per min., or 149,000 cu. ft. per ton of product in 24 hours.

TABLE 1 DISTRIBUTION OF GAS FROM BLAST FURNACE NO. 6

August 1909

	Million B.t.u.	Per Cent
Total gas generated	324.1	100
Stoves and leakage	130.0	40
Blowing engines	92.1	28.4
Used at furnace	9.0	2.8
Auxiliaries	4.6	1.4
Total used for blast furnace operation	235.7	72.6
B.t.u. surplus for furnace	88.4	27.4

B.h.p. equivalent of surplus 1470

3 The results of nitrogen and carbon methods, do not, however, always agree as closely as in this instance. The average coke analysis for the month of August, from which the amount of carbon charged into the furnace was determined, was as follows:

	Per cent moisture	Per cent fixed carbon (dry)
Pocahontas	2.64	90.10
Connellsville	1.94	87.38

4 For the distribution of gas it is assumed that 40 per cent goes to the stoves and is lost by leakage. The amount of gas used for the blowing engines is calculated from the boiler horse power, the latter being determined by the quantity and pressure of air blown, the steam rates of engines and the evaporation per boiler horsepower. The B.t.u. equivalent of boiler horsepower for steam distribution in the current month is equal to $\frac{\text{boiler h. p.} \times 33,320}{\text{boiler efficiency}}$ and the latter is arbitrarily assumed to be 55 per cent. It is further assumed that 150 boiler h.p. is used in steam at each blast furnace and that all auxiliaries use 5 per cent of the steam going to the blowing engines. The difference is the surplus gas per furnace available for other departments, which is stated in the equivalent of boiler horsepower. Using the former example for an illustration, the total amount of B.t.u. in the gas produced per hour by furnace No. 6 was in August 1909, according to the carbon method, 324.1 million. The distribution was as above in Table 1.

5 The surplus gas was used for operating gas engines in the gas electric station and for raising steam for steam electric power. In this way a B.t.u. balance can be made for all furnaces in each month, as the total kilowatt hours produced in the electric stations and the amount of coal which had to be fired under the boilers are known.

TABLES RELATIVE TO BLAST FURNACE GAS

TABLE 2 GAS PRESSURES

MONTHLY AVERAGES, INCHES OF WATER

1909	Jan.	Feb.	March	April	May	June	Jan.-June Average	
Entering gas cleaning plant..	5.0	5.7	7.3	7.6	7.1	7.6	6.7	
After Theisen washers	8.0	8.0	10.1	9.4	9.7	9.8	9.2	
Barometer, inches of mercury	29.44	29.24	29.21	29.36	29.31	29.40	29.33	
	July	Aug.	Sept.	Oct.	Nov.	Dec.	July-Dec.	Jan.-Dec.
Entering gas cleaning plant	9.1	10.0	12.0	13.4	14.6	12.2	11.9	9.29
After Theisen washers.........	11.9	15.0	13.9	16.5	17.7	14.8	14.9	12.10
Barometer, inches of mercury......	29.34	29.37	29.44	29.45	29.42	29.41	29.45	29.37

TABLE 3 COMPOSITION OF GAS FROM INDIVIDUAL FURNACES

AVERAGES

Blast Furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.	Product
1	4.36	33.71	3.41	7.75	120.4	Ferro-Silicon
2	13.47	26.34	4.43	1.95	96.7	Basic
3	14.98	23.97	4.43	1.60	91.4	Basic
4	13.91	24.99	4.10	1.79	94.1	Basic
5	14.17	25.61	3.85	1.81	95.5	Bessemer
6	13.65	25.32	4.26	1.85	95.7	Bessemer

TABLE 4 COMPOSITION OF MIXTURES OF GAS FROM VARIOUS FURNACES

AVERAGES

Blast Furnace No.	CO_2	CO	H	$\frac{CO}{CO_2}$	B.t.u.	Product
1 and 2..............	8.92	30.02	3.92	4.85	108.5	....
1, 2 and 3...........	10.60	28.01	4.09	2.65	102.8	....
2, 3 and 4...........	14.12	25.10	4.32	1.80	94.1	Basic
1, 2, 3 and 4.........	11.68	27.25	4.09	2.33	100.6	....
2 and 3..............	14.23	25.15	4.43	1.81	94.0	Basic
1, 2, 3, 4, 5 and 6....	12.25	26.66	4.08	2.23	98.9	....

TABLE 5 AVERAGE COMPOSITION OF BLAST FURNACE GAS

At 62 Deg. Fahr. and 30 Inches Mercury

	Jan.	Feb.	Mar.	Apr.	May	June	Avg. Jan.-June
CO_2	13.26	13.38	11.53	12.43	13.10	13.20	12.82
CO	25.61	25.50	28.10	26.67	26.56	26.50	26.49
H	2.99	3.95	2.92	3.16	3.74	3.89	3.44
CH_4	0.21	0.23	0.24	0.21	0.22	0.18	0.215
Computed B.t.u	93.04	95.70	100.50	98.81	97.70	98.20	97.32
Ratio $\frac{CO}{CO_2}$	1.93	1.90	2.43	2.15	2.02	2.00	2.07
Heat value per cu. ft. by Calorimeter B.t.u	93.45	96.10	101.00	98.08	98.32	97.99	97.49

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
CO_2	14.10	12.50	10.03	11.96	13.88	13.75	12.53	12.67
CO	25.90	27.30	29.80	26.02	24.70	25.85	26.54	26.51
H	3.86	4.06	3.77	3.45	3.59	3.98	3.78	3.57
CH_4	0.17	0.18	0.19	0.21	0.19	0.15	0.18	0.196
Computed B.t.u	95.70	101.40	108.70	102.90	86.70	95.80	98.40	97.90
Ratio $\frac{CO}{CO_2}$	1.83	2.18	2.98	2.17	1.78	1.88	2.12	2.09
Heat value per cu. ft. by Calorimeter B.t.u	95.50	101.60	107.40	103.10	92.90	94.60	99.20	98.30

APPENDIX NO. 3

DESCRIPTION OF METHODS AND INSTRUMENTS USED IN OBTAINING DATA UPON THE PERFORMANCE OF THE GAS CLEANING PLANT

Temperatures. The temperature of the gas entering the cleaning plant is measured and automatically recorded by a Bristol pyrometer. Readings of the gas temperature are taken by the operators every three hours, after the dry cleaning plant, after the wet scrubbers, and before and after the Theisen washers, the last temperature being also recorded by a Bristol recording thermometer. Temperature readings are further taken at the inlet and outlet of each gas holder. All thermometers have the Fahrenheit scale and are permanently installed in the pipe lines. The water temperatures are read every three hours at the pump suction, after each wet scrubber and after the Theisen washers. For the purpose of comparison the temperature of the atmosphere near the washer building is simultaneously recorded. Portable thermometers with the Fahrenheit scale are used for measuring water temperatures.

2 *Pressures.* Readings are taken by the operators every three hours, by means of ordinary U-tubes, of the gas pressure in inches of water at the following places: at the point where the gas enters the cleaning plant, between the *wet* scrubbers, and before and after the Theisen washers. In addition, the pressure of the raw gas at the main water seal, and of the fine gas after the Theisen washers, is continuously recorded by Bristol pressure gages. For convenience of observation all instruments indicating and recording gas and water pressures, temperatures and venturi meter pressure differences, are arranged on a *gage* board (Fig. 1)installed in the Theisen washer building. On the same gageboard the telephone gongs and the optical indicators showing the position of the two gas-holder bells are mounted. By a simple system of contacts each gas-holder bell, while descending, closes five different electric circuits, and causes incandescent lights to burn corresponding to its different positions. Thus a white light burns when the holder is in its top position; a green light appears when the holder bell has descended 7 ft.; one red light, indicating danger, corresponds to a 14-ft. immersion of the holder bell, and two, and at last three, red lamps show that the bell has fallen 21 ft. and 28 ft. respectively, and is nearing its bottom position. When the three red lights burn the gas holder is practically empty. Similar optical indicators are installed in the electric power and gas blowing-engine houses, so that the engine operators can independently observe the position of the gas holders at any time.

3 *Power Consumption.* As all Theisen washer and pump motors are operated by 440-volts alternating current reduced in special transfomers from 2200 volt, 25 cycle, 3 phase current generated in the electric station, an integrating kilo-

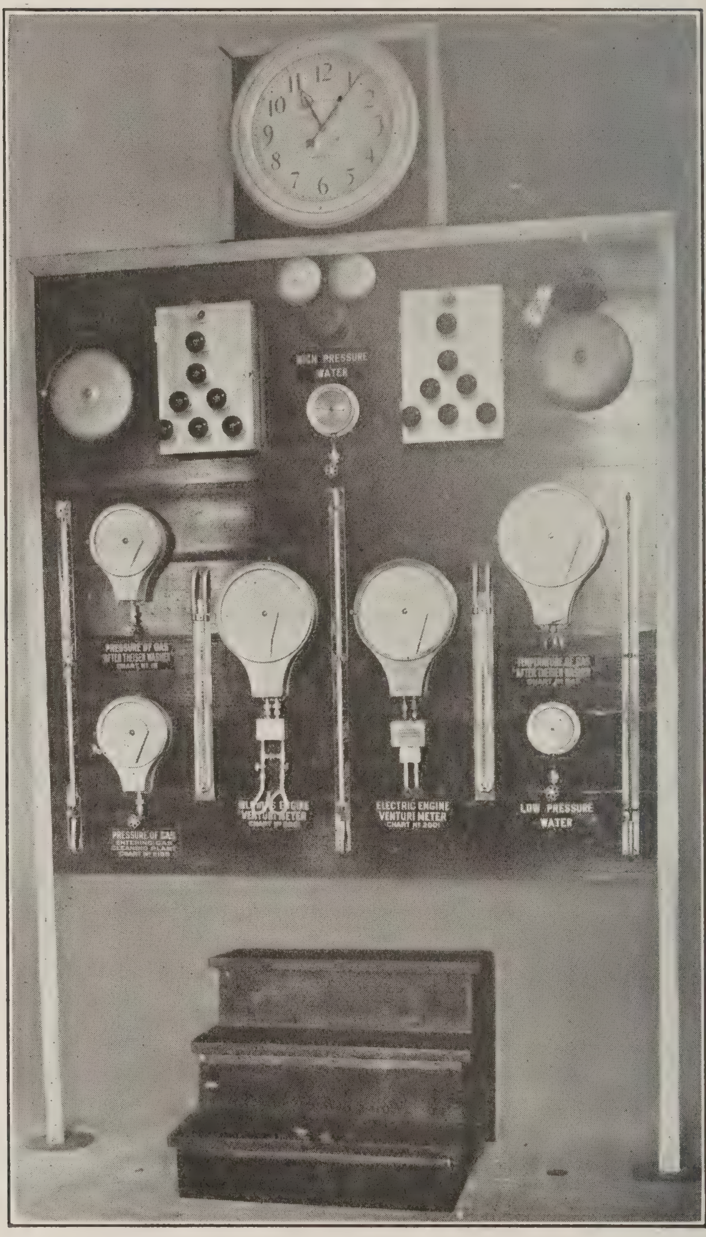

FIG. 1 GAGEBOARD IN THEISEN WASHER BUILDING.

watt meter was installed at the main switchboard to measure the combined power consumed by all gas cleaning plant motors. Unfortunately, the motor-driven air compressors for starting the gas engines are on the same line, so that the power for two 50-h.p. motors is included in the meter readings; but since each air compressor is running only about one hour each day, the error is believed to be of little weight. However, it must be kept in mind that on this account the recorded power consumption of pumps and Theisen washers is higher than the actual values. A portable ampere meter for use in the Theisen washer building can easily be attached to each motor and read by the operator every three hours, and the readings are recorded in the daily report sheets. These ampere meter readings give the only indication of the load carried by each Theisen washer. As all washers operate in parallel, one machine, by a slight misadjustment of the inlet and outlet gate valves, may handle more gas, thus carrying more load, and cleaning its share of the gas to a less extent than the other washers. The ampere meter will indicate such an inequality in the load distribution, and the operators have orders to keep the ampere readings on all washers in operation fairly uniform, by proper adjustment of the gate valves. The indications of the ampere meter, averaged for each month, are used to divide the power charges in proper proportion between the pumps in the preliminary washing plant and the Theisen washers.

4 *Water Consumption.* The amount of water consumed each minute in the gas-washing plant is measured by overflow weirs at the settling tank. A hook gage gives directly the number of gallons of water per minute falling over the weir so that the operator reads the water quantity as easily as a thermometer. The settling tank has two compartments, so that the water from the wet scrubbers can be turned into one, while the Theisen waste water is flowing into the other compartment. Gates make it possible to reverse these flows and to shut off each settling tank for cleaning. The amount of water allowed at each wet scrubber and Theisen washer, originally apportioned by means of calibrated barrels, is adjusted in practice by estimating the relative quantity basing the estimate on the thickness of the stream at each Theisen water inlet and at the wet scrubber overflow. The water used for gas-washing purposes is waste cooling water from the blast furnaces, formerly discharged into the sewer but now collected and piped to the Theisen washers under its natural head, while another part is lifted on top of the wet scrubbers by two centrifugal pumps of 2,000,000-gal. capacity each. These pumps receive the water under 30 ft. head and deliver it at a pressure of 80 ft. for distribution through the sprinkler system.

5 *Dust and Moisture.* Gas engine builders usually specify the amounts of dust and moisture which should not be exceeded for safe operation. Recent specifications call for blast furnace gas containing not more than 0.02 grains of flue dust and not over 10 grains of moisture per cu. ft., at 62 deg. fahr. and 30 in. mercury. With an efficient, modern gas-cleaning plant it is not difficult to meet these conditions, as is shown in the paper. Dust and moisture determinations are made in the gas laboratory, and recorded on daily chemical report blanks (Fig. 6 in the body of the paper). The amount of dust is determined in dry cleaned gas, clean gas and fine gas, and occasionally in the atmosphere, while moisture determinations are made in clean and in fine gas. For purposes of comparison the moisture in the atmosphere, as determined at the dry-blast

plant, is also recorded. Results are given in grains per cubic foot of standard gas, and by a simple calculation the efficiency of the wet scrubbers and of the secondary washing plant can be determined each day.

6 While frequently tried, it has been found impossible to make dust determinations in raw gas, which give more than a general idea of the efficiency of the dry cleaning plant. The difficulty is that the raw gas frequently carries larger particles of dirt, which obstruct the sample pipe and clog the dust filter before a gas sample of sufficient size for correct determination can be secured.

7 A series of tests extending over eight days was made in August 1908, with the regular dust filter apparatus then in use, but since no results were obtainable, a long glass tube open at both ends and filled with dry calcium chloride was then weighed and attached to the gas main in place of the dust filter. After passing several cubic feet of gas, the glass tube was taken off, closed at both ends and weighed, the difference in weight giving the total amount of dust and moisture. The amount of dust was determined by drying and weighing again as the difference between the second and third weighings. This apparatus was found to be a little more satisfactory, as larger samples could be taken, but gradual filling of the pipe with dust made the accuracy of the results very doubtful. The size of the gas sample secured never exceeded 6 cu. ft., with an average of about 2 cu. ft.

8 Tests made in August 1907 gave similarly unreliable results, as the amount of dust in raw gas, according to these determinations, varied from 0.18 grains per cu.ft.at 2 p.m., August 7, to 563.19 grains per cu. ft.at 4.30 p.m. August 29. Such extreme variations are improbable, and as the results obtained cannot represent a fair average, raw gas dust tests were discontinued as of questionable value.

9 The method and the instrument used for the determination of dust in clean and fine gas were developed by Messrs. Wm. Brady and L. A. Touzalin. The Brady filter (Fig. 2) consists essentially of a brass cylinder provided with inlet and outlet and supporting the filter itself, which is an ordinary 94 x 33 mm. Soxhlet extraction shell. The brass cylinder is of such diameter that an annular space of about $\frac{3}{16}$ in. is formed between its inner surface and the paper filter. The filtering shell is fastened and held tightly in place without the aid of gaskets, by wedging its open end between the tapering cylindrical faces of the brass shell and the brass nozzle, as shown in the illustration. This method of fastening has the additional advantage that for a distance of about one-half an inch from the edge, the Soxhlet shell is protected from dust deposits so that the filter can safely be handled after the experiment. The brass nozzle forming the inlet of the apparatus is provided with inside threads so that it can be screwed to a sampling pipe. Its inside surface is perfectly smooth, without ledges or places for the accumulation of dust. A brass nut holds the three parts of the instrument in place. The apparatus may be used in any position, but the preferred arrangement is horizontal.

10 When the gas to be filtered contains moisture the filtering device must be heated to about 110 deg. cent. by any suitable means, but preferably by surrounding the brass shell with an electrically heated sleeve as shown in Fig. 3, which represents the sampling pipe, Brady filter, moisture tubes and gas meter assembled ready for use. The nipple on the outlet end of the brass cylinder is threaded so that it can be removed and replaced by aluminum

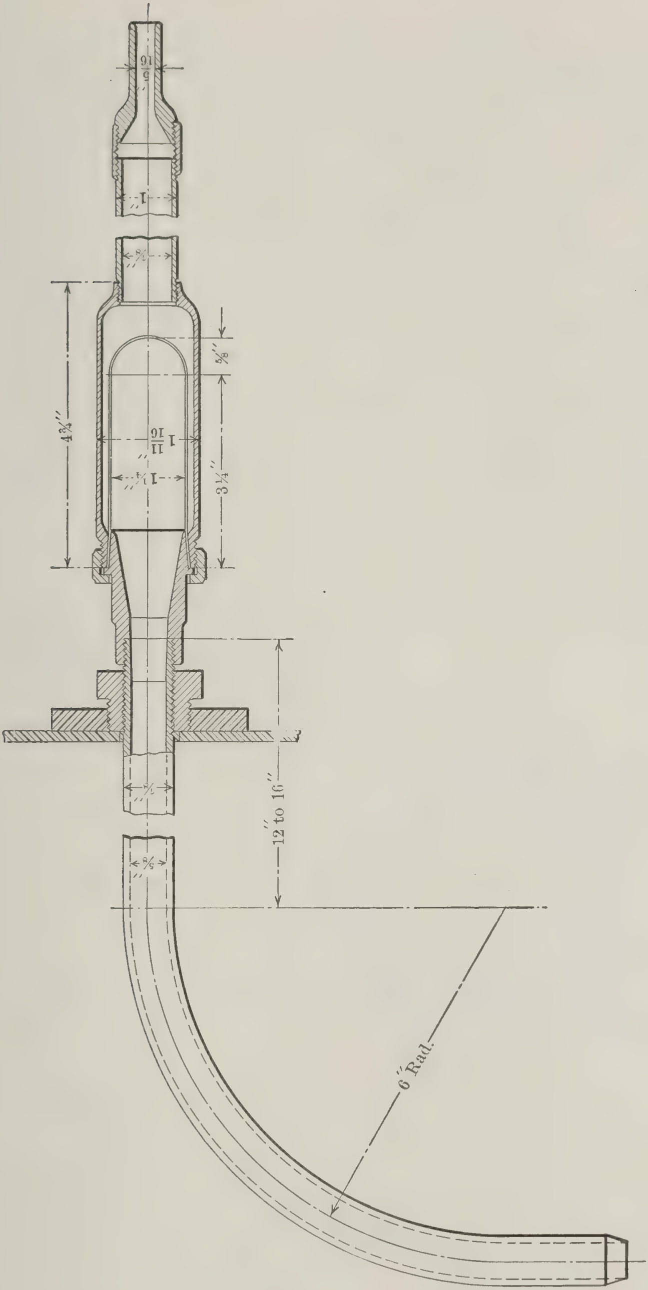

FIG. 2 SECTION OF BRADY DUST FILTER

tubes filled with a dehydrating agent such as calcium chlorid, in case it is desired to determine the moisture in the gas.

11 The Brady dust filter was chosen for use as standard instrument on account of its advantages over other methods. It is very simple, easily assembled and taken apart, perfectly tight, and it fulfills the requirement that the gas issuing from the sample pipe should pass over very little surface before being filtered. The principal advantage, however, is the use of a strong cylindrical filter which resists the gas pressure much better than the thin sheet of filter paper used in other instruments. The Soxhlet shell maintains a porous condition even though much fine dust has been deposited owing to the formation of concentric layers of dust and their subsequent cracking by the action of gravity and slight jarring. Gas samples of twice the size permissible in other instruments can be passed through the Brady filter. Fig. 4 shows five Brady filter shells after use for determination of dust at the main water seal, before and after the wet scrubbers, after the Theisen washers, and after the power station gas holder. The feature of keeping the filter porous for large samples is noticeable, particularly in the shell on the extreme left.

12 Whenever determinations of dust and moisture, or both, are desired, the instrument must be placed as close as possible to the pipe where the sample is to be taken. The sampling pipe used, shown in Fig. 3, consists of a ½-in. brass pipe curved with a radius of not less than 6 in. and smoothly polished on the inside. It is inserted in the gas flue, if possible on a horizontal diameter at least 15 ft. away from any bend or obstruction, to a distance of one-fourth to one-third of the diameter. The inlet opening is reduced to a sharp edge, so that there is as little local disturbance at that point as possible. The question of the proper form of sampling pipe was decided in favor of the curved pipe, against the straight pipe with standard 4-in. insertion in the gas main. Experiments were made at various times to determine the amount of flue dust, by simultaneously using both forms of sampling pipe inserted at practically the same place in the gas flues.

13 A comparison of the results of these tests shows plainly the difference in the effect of straight and curved sampling pipes on the size of the gas sample, which generally speaking is larger with the curved pipe. This advantage is, however, of secondary importance, compared with the material increase in the dust contents recorded by sampling the gas with curved pipes. This increase is particularly noticeable in testing dry cleaned gas, and shows that the heavier particles of dust cannot easily be induced to change their direction of travel to enter the straight sample pipe at right angles, but pass by the opening, which is parallel with the gas stream. The difference in results averages nearly 100 per cent in favor of the curved sampling pipe.

14 In the clean gas tests this difference is considerably less marked and the average size of the gas sample obtained is even smaller, while the amount of dust recorded when using the curved sampling pipe is only 8½ per cent larger, because the dust remaining in clean gas is of much finer quality and of less specific gravity, so that the particles will much more easily change their direction of travel. No difference at all, however, can be observed when dealing with fine gas. The averages of two series of tests coincide exactly, and only slight deviations are noticeable in the individual readings. The average size of the gas sample taken with the curved pipe is 14 per cent larger then the sample

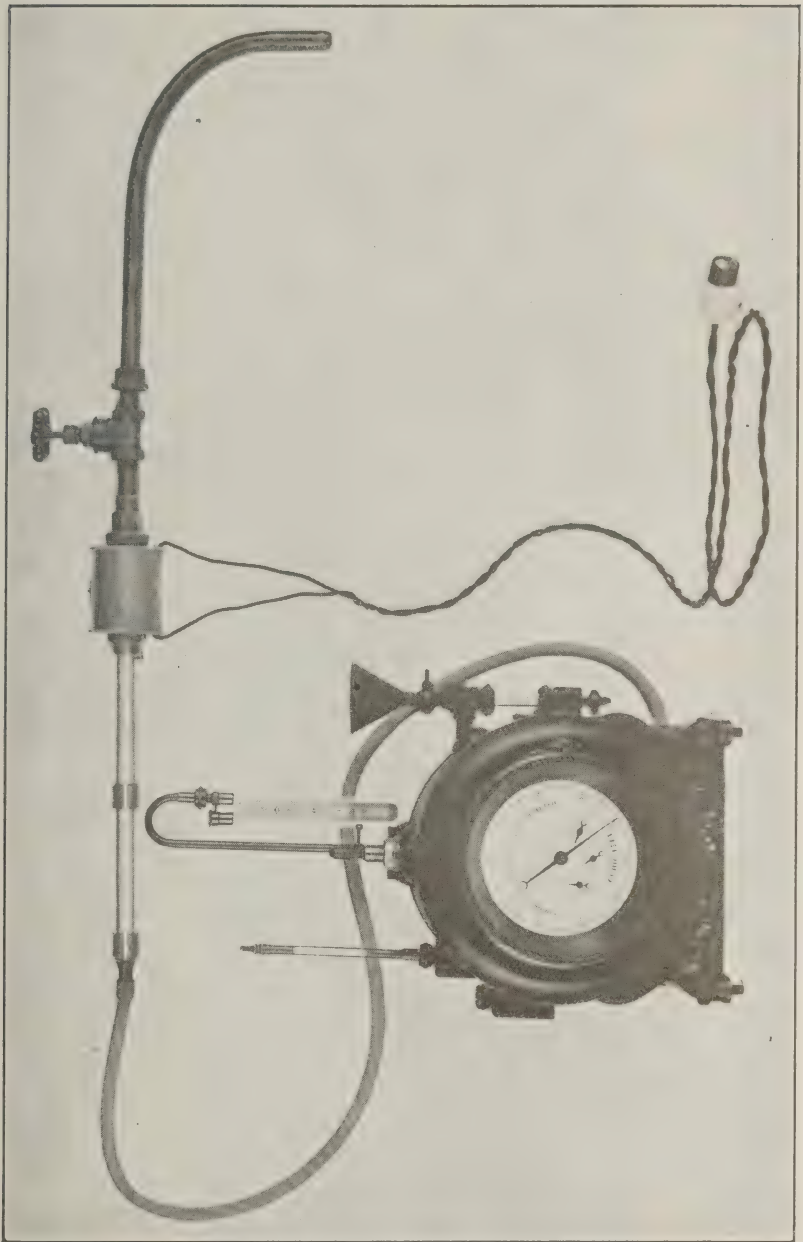

Fig. 3 Brady Filter, Sampling Pipe and Gas Meter

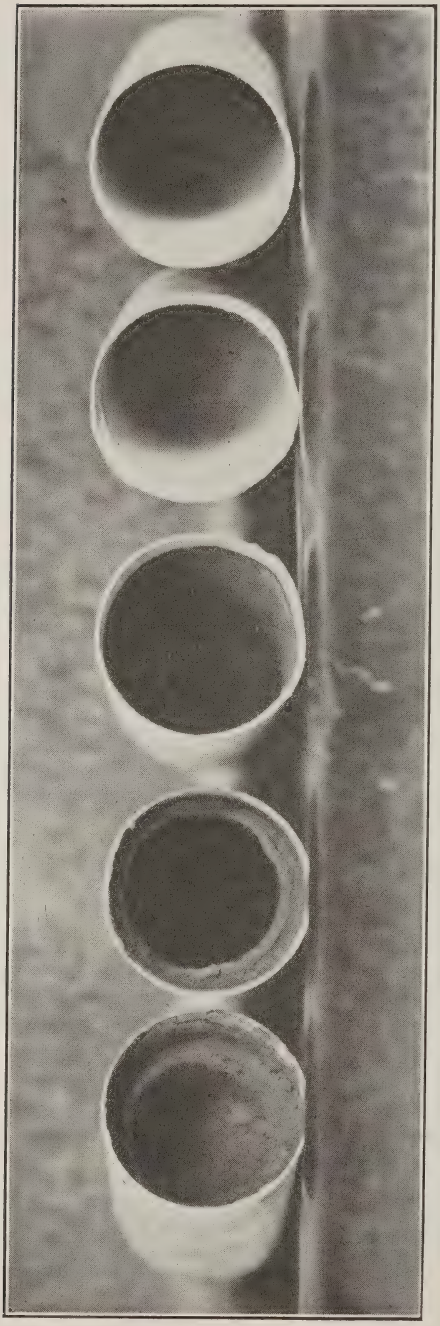

Fig. 4 Brady Filter Shells After Use

obtained with the straight sampling pipe. This experience is valuable as it shows that dust determinations in fine gas are very much less influenced by variations in the method of sampling, which further suggests that results will not be materially affected by variations in the instruments and methods used for dust determinations in fine gas.

15 As the original gas pressure is usually—in fine gas always— sufficient to cause the gas to flow through the filter, an aspiration of the sample is unnecessary. The determination is made as follows: The Soxhlet shell is first dried and weighed in a glass weighing bottle, then inserted into the brass shell of the apparatus, and the parts tightly connected. A meter reading is taken after the instrument is connected to the sample pipe and the gas is turned on. The dust contained in the gas sample is deposited in the Soxhlet shell, while the moisture is driven over into the aluminum tubes containing anhydrous calcium chlorid, and connected in series. They are capped and weighed before the experiment and their increase in weight represents the moisture in the volume of gas passed through the apparatus, while the increase in weight of the shell after drying gives the amount of dust carried in the gas sample. For dust determinations from 30 to over 200 cu.ft. of gas are passed, depending on conditions of pressure and locality, while for moisture determinations only from one to five cu. ft. are used. Usually one or several moisture tests may be made while one dust test is being run, simply turning off the gas for a moment when the aluminum tubes are inserted and again when they are withdrawn. Readings of the meter and of the gas conditions such as temperature and pressure must of course be made at the beginning and end of each test. All results of dust and moisture tests are calculated to grains per cubic feet of standard gas.

16 The application of two Brady filters permits continuous determination, a feature of great importance in gas power plants since it permits uninterrupted surveillance of the gas cleaning plant and of its efficiency. Continuous dust determinations are being made every day except Sunday, by sampling dry cleaned gas, clean gas and engine gas. A Brady filter is started at each place at 8.30 a.m., and the gas is allowed to pass until 4.30 p.m., when the Soxhlet shell is removed and a new one inserted, which is in continuous use from 5.00 p.m. until 8.00 a.m. the following morning. The average size of sample for day and night runs respectively, is from 60 to 90 cu.ft. of dry cleaned gas, 80 to 160 cu.ft. of clean gas, and 120 to 200 cu.ft. of engine gas. It is evident that such large accumulative samples must very nearly represent a true average of the amount of dust contained in the gas. Occasional dust and moisture determinations usually practiced in the majority of plants are of comparatively little value, as they do not give the true average conditions of the gas. Comparisons of results obtained at different gas-cleaning plants cannot and should not be made and credited, unless all instruments, methods, size of samples, duration of tests, etc., are identically the same. Standardization of the method of determining dust and moisture in industrial gases would benefit the gas engine industry at large, and the method used at this plant, which has been thoroughly and continuously tried under all conditions, is worthy of consideration as a basis for standardization.

APPENDIX NO. 4

RESULTS IN DETAIL OF OPERATION OF GAS-CLEANING PLANT

TABLE 1 HEAT LOSS OF GAS BY RADIATION

MONTHLY AVERAGES

1908	July	Aug.	Sept.	Oct.	Nov.	Dec.	Average
Gas cleaned, cu. ft. per min.	14,020	12,850	16,950	18,070	17,690	13,090	15,420
Temperature of gas at main water seal, deg. fahr....	426	410	303	312	299	329	346
Temperature of gas to scrubber No. 1, deg. fahr.....	210	196	202	168	150	133	177
Difference:							
Loss by radiation, deg. fahr....................	216	214	101	144	149	196	169
Reduction in per cent...	50.7	52.2	33.3	46.1	49.8	59.5	48.8

TABLE 2 WET SCRUBBER EFFICIENCY

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.–June	
Flue dust in dry cleaned gas, gr. per cu. ft......................	0.4772	0.4787	1.2951	1.0335	1.1172	1.0804	0.9137	
Flue dust in clean gas, gr.per cu.ft	0.0766	0.1224	0.2238	0.2178	0.2146	0.2389	0.1825	
Difference........................	0.4006	0.3563	1.0713	0.8157	0.9026	0.8415	0.7312	
Per cent removed by wet scrubbers	84.0	74.5	82.8	79.0	80.8	77.8	79.7	
	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Flue dust in dry cleaned gas, gr. per cu. ft.......................	0.7990	1.6124	4.0940	2.8794	2.4004	1.1188	2.1506	1.5330
Flue dust in clean gas, gr. per cu. ft...............................	0.1316	0.3669	0.8257	0.9882	0.2539	0.1786	0.4541	0.3183
Difference.......................	0.6674	1.2455	3.2683	1.8912	2.1465	0.9402	1.6965	1.2147
Per cent removed by wet scrubbers............................	83.5	77.4	79.9	65.7	89.6	84.2	78.8	79.3

TABLE 3 GAS AND WATER TEMPERATURES

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.-June
Temperature of gas to wet scrubber No. 1, deg. fahr	123.20	126.00	161.00	152.00	163.20	172.00	149.60
Temperature of gas to Theisen washers, deg. fahr	41.40	44.00	49.60	55.50	63.80	74.20	54.70
Temperature of gas to gas holder, deg. fahr	40.10	43.00	44.10	52.50	64.20	69.70	52.30
Temperature of water supply, deg. fahr	43.80	45.00	44.20	52.30	62.80	71.70	53.30
Waste, wet scrubber No. 1, deg. fahr	59.20	63.00	62.50	67.20	77.00	84.80	68.90
Waste, wet scrubber No. 2, deg. fahr	43.30	44.00	46.80	52.70	63.20	72.90	53.80
Temperature of air, deg. fahr	27.90	31.00	37.10	46.40	56.90	65.10	44.10

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Temperature of gas to wet scrubber No. 1, deg. fahr	169.20	173.60	150.60	159.50	119.30	86.70	143.30	145.70
Temperature of gas to Theisen washers, deg. fahr	76.50	79.10	72.70	61.80	61.70	52.80	67.40	61.10
Temperature of gas to gas holder, deg. fahr	79.10	80.90	72.30	61.00	61.10	42.00	54.40	56.60
Temperature of water supply, deg. fahr	74.70	78.60	72.80	63.70	64.50	46.20	66.70	60.00
Waste, wet scrubber, No. 1, deg. fahr	89.00	96.70	95.50	94.70	92.50	88.30	92.80	80.80
Waste, wet scrubber, No. 2, deg. fahr	76.70	81.50	77.10	64.90	65.40	52.70	67.70	61.70
Temperature of air, deg. fahr	83.40	85.20	74.60	60.80	58.80	31.20	65.60	54.90

TABLE 4 EFFICIENCY OF SECONDARY WASHING PLANT

MONTHLY AVERAGES

1909	Jan.	Feb.	March	April	May	June	Avg. Jan.-June
Flue dust in clean gas, gr. per cu. ft	0.0766	0.1224	0.2238	0.2178	0.2146	0.2389	0.1825
Flue dust in fine gas, gr. per cu. ft.	0.0036	0.0057	0.0044	0.0059	0.0067	0.0067	0.0055
Difference	0.0730	0.1167	0.2194	0.2119	0.2079	0.2322	0.1770
Per cent removed by refining	95.4	95.4	98.0	97.4	96.6	97.3	97.0

TABLE 4—CONTINUED.

	July	Aug.	Sept.	Oct.	Nov.	Dec.	Avg. July-Dec.	Avg. Jan.-Dec.
Flue dust in clean gas, gr. per cu. ft.	0.1316	0.3669	0.8257	0.9882	0.2539	0.1786	0.4541	0.3183
Flue dust in fine gas, gr. per cu. ft.	0.0057	0.0058	0.0080	0.0074	0.0069	0.0093	0.0061	0.0058
Difference	0.1259	0.3611	0.8177	0.9808	0.2470	0.1693	0.4480	0.3125
Per cent removed by refining	95.6	98.5	99.0	99.1	97.5	95.0	98.7	98.1

TABLE 5 DETERMINATION OF FLUE DUST AT DIFFERENT POINTS OF SECONDARY CLEANING PLANT

GRAINS PER CUBIC FOOT

1910	Turn	After wet scrubbers	Before Theisen washers	After Theisen washers	After gas holder
March 21	night	0.1496	0.1091	0.0024	0.0024
March 22	day	0.1681	0.1468	0.0066	0.0051
March 22	night	0.1647	0.1292	0.0045	0.0042
March 23	day	0.1773	0.1563	0.0091	0.0094
March 23	night	0.1185	0.1145	0.0062	0.0065
March 24	day	0.1566	0.1489	0.0058	0.0058
March 24	night	0.1456	0.1371	0.0066	0.0064
March 25	day	0.1568	0.0983	0.0067	0.0061
March 25	night	0.1474	0.1258	0.0064	0.0057
March 26	day	0.1425	0.1078	0.0037	0.0029
Average		0.1527	0.1274	0.0058	0.00545

Average amount removed
- By clean gas main .... 0.0253 grains per cu. ft.
- By Theisen washers .... 0.1216 grains per cu. ft.
- By fine gas main and gas holder .... 0.00035 grains per cu. ft.

Average absolute efficiency of
- Clean gas main .... 16.56 per cent.
- Theisen washers .... 95.45 per cent.
- Fine gas main and holder .... 6.03 per cent.

The average total efficiency of the secondary cleaning plant was 96.43 per cent, in which the three above factors participated as follows:
- Clean gas main .... 16.56 per cent.
- Theisen washers .... 79.64 per cent.
- Fine gas main and gas holder .... 0.23 per cent.

Total .... 96.43 per cent.

TABLE 6 DUST IN COMBUSTION AIR

Date	Wind	Day Turn: Temp of Air Deg. Fahr.	Day Turn: Barometer	Day Turn: No. cu. ft. sample	Day Turn: Dirt grams per cu.	Weather	Night Turn: Temp. of Air	Night Turn: Barometer	Night Turn: No. cu ft. sample	Night Turn: Dirt grams per cu. Ft.
7-12-09	W.	78	29.21	74.13	0.0052	Part Cloudy	70	29.31	81.86	0.0043
7-13-09	W.	76	29.19	95.57	0.0032	Part Cloudy	71	29.26	57.59	0.0037
7-14-09	S. W.	80	29.27	124.51	0.0013	Wind blowing hard	73	29.28	102.84	0.0052
7-15-09	W.	84	29.28	126.41	0.0048	Part cloudy	76	29.28	129.11	0.0036
7-16-09	N. W.	80	29.34	88.69	0.0004	Wind hard	86	29.40	95.46	0.0005
7-17-09	N. W.	84	29.39	145.82	0.0004	Clear	..	......	......	....
Average	...	......	......	109.19	0.00255	....	.....	....	93.37	0.00346

TABLE 7 ANALYSES OF DUST DEPOSIT ON GAS AND AIR DAMPERS OF GAS ENGINE NO. 1

Sample February 1909	Gas Damper 1	Gas Damper 2	Air Damper 3	Air Damper 4
Silica	19.60%	22.50%	32.40%	23.80%
Alumina	12.07	20.19	6.50	11.30
Iron	6.95	6.37	11.12	12.03
Manganese	2.52	2.62	1.04	1.15
Lime	32.74	23.00	5.84	5.37
Magnesia	3.43	3.38	0.92	0.95
Volatile	17.89	17.38	37.84	39.85

TABLE 8 ANALYSES OF FLUE DUST REMOVED FROM BLAST FURNACE GAS AT DIFFERENT STAGES OF GAS CLEANING

March 1908	SiO_2	Al_2O_3	Fe	CaO	MgO	Fix C.	Sul.	Phos.	Mang.	Vo.
From main water seal (deposit)	10.30	4.60	48.85	2.46	0.30	15.52	....	0.067	0.53	....
From collecting main after dry cleaning plant (deposit)	11.44	4.45	42.86	3.15	0.68	10.28	0.202	0.079	0.80	7.28
From No. 1, wet scrubber (sediment)	14.58	5.45	43.09	2.75	0.78	9.17	0.288	0.095	0.60	4.17
From No. 2, wet scrubber (sediment)	18.26	5.85	38.20	4.74	1.40	8.54	0.192	0.097	0.47	5.08
From Theisen washers (sediment)	22.93	7.94	26.06	7.60	1.61	11.47	0.314	0.119	1.08	6.73

TABLE 9 ANALYSIS OF FLUE DUST REMAINING IN BLAST FURNACE GAS AT DIFFERENT STAGES OF GAS CLEANING

MARCH 1909

Location of Brady filters	SiO_2	Al_2O_3	Fe_2O_3	CaO	MgO	Mn.	Vol.
At main water seal	12.19%	5.93%	52.39%	4.70%	0.97%	1.16%	22.32%
Gas entering wet scrubber No. 1	11.37	5.21	52.55	3.76	0.86	0.83	25.18
In clean gas main	21.14	11.53	28.35	9.56	1.94	2.46	24.31

TESTS ON WET SCRUBBERS

1 Tests were made on October 27, 1908, to determine the cooling and condensing effect of the wet scrubbers. The temperature of the water and gas entering and leaving the washers were taken with accurate thermometers.

TABLE 10 WET SCRUBBER TEST

Time	Temperatures							Quantities			
	Water				Gas			Gas	Water		
	Scrubbers				Scrubbers			cu. ft. per min.	gal. per min.		
	Inlet 1 and 2	Outlet 1	Outlet 2	Outlet 1 and 2	Inlet 1	Outlet 1	Outlet 2		No. 1	No. 2	Total
10.50	63.0	80.0	64.0	73.0	157.0	62.8	60.0	17,000	.....	.....	1,380
11.00	62.8	77.5	63.0	71.5	156.0	62.8	60.0	17,150	.....	.....	1,375
11.10	63.0	77.5	63.2	72.0	156.0	62.8	60.0	17,350	.....	.....	1,375
11.20	63.0	76.5	63.5	71.5	155.0	62.8	60.0	17,150	.....	.....	1,375
11.30	62.8	77.2	63.0	71.5	155.0	62.8	60.0	17,150	.....	.....	1,375
Average....	62.9	77.74	63.34	71.9	155.8	62.8	60.0	17,160	816	560	1,376
2.55	63.6	85.5	63.0	77.0	180.0	64.8	62.0	23,100	.....	.....	1,225
3.05	63.8	85.5	63.0	76.5	177.0	64.8	62.0	23,400	.....	.....	1,275
3.15	63.8	85.5	63.0	76.0	175.0	64.8	61.5	23,100	.....	.....	1,225
3.25	63.5	85.0	63.0	76.5	175.0	64.3	61.5	23,250	.....	.....	1,290
3.35	63.5	87.0	63.2	77.0	176.0	64.3	61.5	23,100	.....	.....	1,260
Average....	63.64	85.7	63.04	76.6	176.6	64.5	61.7	23,190	752	503	1,255

The total amount of water from both washers was measured by the weir, and the amounts passing through each washer were calculated from the final temperatures. The gas was measured by venturi meter. The temperature of the atmosphere was 45 deg. fahr. Two tests of 40 minutes each were made on the same day. The readings and averages are given in the table.

2 The total heat absorbed in the first scrubber during the first test was 100,912 B.t.u. per min., of which 31,205 B.t.u. is accounted for in the loss of sensible heat in the gas. On the second test the total heat absorbed was 137,874 B.t.u. per min., of which 50,848 B.t.u. is accounted for by the loss of sensible heat. From the following calculations the amount of vapor condensed per cu. ft. of gas at 64 deg. was found to be 27.4 grains in the first, and 25.2 graids in the second test, or an average of 26.3 grains.

First Test:

Washer 1, temperature of entering gas, 155.8 deg.; leaving gas, 62.8 deg.

Density of gas = 0.0815

Specific heat = 0.24

Cu.ft. of gas per min. = 17.160

Sensible heat lost by gas:

$17,160 \times 0.0815 \times 0.24 \times (155.8 - 62.8) = 31,205$ B.t.u. per min.

Heat absorbed by water:

$816 \times 8\frac{1}{3} \times (77.74 - 62.90) = 100,912$ B.t.u. per min.

100,912 B.t.u. − 31,205 B.t.u. = 69,707 B.t.u.

Average latent heat of vapor from 155.8 deg. to 62.8 = 1037 B.t.u.

$\frac{69,707}{17,160 \times 1037} = 0.00392$ lb. or 27.4 gr. of vapor condensed per cu. ft. of gas at 64 deg.

Second Test:

Sensible heat lost by gas:

$23,190 \times 0.0815 \times 0.24 \times 112.1 = 50,848$ B.t.u.

Heat absorbed by water:

$752 \times 8\frac{1}{3} \times 22.06 = 137,874$ B.t.u.

173,874 B.t.u. − 50,848 B.t.u. = 87,026 B.t.u.

Average latent heat from 176.6 to 64.5 deg. = 1029 B.t.u.

$\frac{87,026}{23,190 \times 1029.7} = 0.003605$ lb. or 25.2 gr. vapor condensed per cu. ft. gas at 64 deg.

3 Since the average temperature of the gas leaving the first scrubber was about 64 deg., the amount of moisture remaining in the gas was about 6.6 grains; this added to 26.3 grains condensed gives 32.9 total grains of moisture per cu. ft. in dry cleaned gas. This represents a dewpoint of about 117 deg. or about 31 per cent saturation at the average initial temperature of 166 deg. Later tests with wet and dry bulb thermometers in the gas mains showed dewpoints varying from 104 deg. to 114 deg. for an average gas temperature of 170 deg. The results of these tests indicate the reducing effect which the washing of the gas has on the moisture. While by these calculations the amount of moisture in the dry cleaned gas was found to be about 33 grains per cu. ft. in October 1908, moisture determinations with Brady filters made on July 14, 15 and 16, 1909, gave very similar results. It will be noted that the test figures fairly coincide with the calculated values.

1908, moisture determinations with Brady filters made on July 14, 15 and 16, 1909, gave very similar results. It will be noted that the test figures fairly coincide with the calculated values.

TABLE 11 MOISTURE TEST IN DRY CLEANED GAS

Date	CO_2	CO	H	CH_4	B. t. u	$\frac{CO}{CO_2}$
July 14 Gas analysis	14.8	25.5	3.0	0.1	91.8	1.72
Grains of moisture per cu. ft.						34.124
Temperature at meter, deg. fahr.						84
Barometer, inches of mercury						29.27
Temperature in gas main, deg. fahr.						192
Pressure in gas main, inches of water						9.5
July 15 Gas analysis	13.8	25.7	4.0	0.1	95.3	1.86
Grains of moisture per cu. ft.						41.7453
Temperature at meter, deg fahr						85
Barometer, inches of mercury						29.68
Temperature in gas main, deg. fahr.						156
Pressure in gas main, inches of water						6.5
July 16 Gas analysis	13.1	26.1	3.5	0.2	96.1	1.99
Grains of moisture per cu. ft.						38.421
Temperature at meter, deg. fahr.						83
Barometer, inches of mercury						29.36
Temperature in gas main, deg. fahr.						190
Pressure in gas main, inches of water						10
Average moisture, gr. per cu. ft.						38.1

APPENDIX NO. 5

METHOD USED FOR MEASURING AND RECORDING GAS CONSUMPTION

The following description of the method used for measuring and recording the gas consumption was contributed by C. J. Bacon, Mem. Am. Soc. M. E.

2 The amount of gas consumed by the blowing engines at the blast furnaces and the power engines in the electric station, is measured by venturi meters one in the 54-in. main to the blowing engines and another in the 60-in. main to the power engines. The 60-in. meter was installed first and tested by volumetric measurements as hereinafter described. The 54-in. meter was subsequently constructed with the same proportions, and as the only difference is in the size no tests have been thought necessary. These meters are of much the usual form, except that certain liberties were taken in the design to simplify the shop work; the throat section of each being a straight cylinder connected to the small ends of the upstream and downstream cones without rounding at the intersections; and there was a similar omission of curvature at the connection between the approach section of 5ft. pipe and the large end of the upstream cone. Although it was realized that these departures from theoretically perfect design were likely to introduce more or less error due to eddy currents, nevertheless in view of the facility with which the accuracy could be determined by means of the gas holder, the somewhat irregular construction was allowed to stand. The absence of test data on meters of this size made tests advisable regardless of how nearly perfect the shape and construction might be.

3 By referring to Fig. 23*b* of the paper, it will be seen that the 60-in. meter has an over-all length of 53 ft. 1 in., and consists of an up-stream cone 11 ft. 6 in. long and having openings 60 in. and 20 in. in diameter, a straight cylindrical throat section of cast iron 20 in. in diameter by 15 in. long, and a downstream cone 39 ft. long, likewise with openings 60 in. and 20 in. in diameter. The up-stream cones are made of plate, with butt-joints and countersunk rivets inside to reduce friction. A cylindrical casting 16 in. long by 60 in. in diameter and containing an annular pressure chamber, is inserted between the straight-approach pipe and the upstream cone. A similar pressure chamber surrounds the throat. Twelve 3/16-in. holes communicate to each of the chambers the pressures existing within the meter at those points. The characteristic equation for flow of gas in venturi tubes[1] was used in the calibration of this meter.

4 A number of carefully conducted tests have been made at various times to determine the meter coefficient, utilizing the 100,000 cu. ft. gas holder as a means of volumetric measurement. This holder is located about 260 ft. from

[1]See The Flow of Fluids in a Venturi Tube, by E. P. Coleman, Transactions, vol. 28, 1907 p. 483, for the derivation of this equation.

the meter, and is provided with a combination of water-sealed valves such that the flow of gas to and from the holder may be controlled at will. The horizontal area of the holder was accurately determined by measurement of diameters, and a vertical scale of feet and tenths was marked on the outside to permit of determination of the rate of rise of the holder. Observations were taken of

- *A* gas pressure in the upstream chamber.
- *B* difference in pressure between upstream and throat chambers.
- *C* temperature of gas at meter and holder.
- *D* analysis of gas including water vapor contents.
- *E* barometric pressure.
- *F* gas pressure at inlet to holder.

From these data and the dimensions of the meter, values may be assigned in the above-mentioned equation of flow. It is worthy of especial note that the ratio of specific heats for the mixture of gas and aqueous vapor is in this case 1.38, the use of it in the formulæ, however, does not result in an appreciably lesser flow than the use of 1.408, the commonly accepted value for air. Without burdening this paper with the actual data and computations, the net average results of 17 separate holder tests at various rates of flow shows a meter coefficient of 0.91, which is taken to mean that the actual flow is 91 per cent of the theoretical flow.

5 This determination of meter coefficient was made more as a matter of scientific interest than as a necessity, since working curves showing the relation between the difference in upstream and throat pressure and volume of gas at prevailing temperatures and pressures, could have been constructed from test data alone. The meter coefficient, however, being available, it was made use of in connection with the theoretical formula in preparing the curves in Fig. 35 of the paper, from which the volume of gas, reduced to standard conditions of 62 deg. fahr. and 29.92 in. barometer, may be determined for prevailing temperatures and meter readings. For these curves the absolute pressure of gas in the main is taken as 29.5 in. mercury, which is the sum of the average upstream pressure and the average barometer at this locality.

6 Daily records of flow of gas are obtained by means of a Bristol differential pressure recorder, located in the Theisen washer building about 250 ft. away from the meter and connected to upstream and throat chambers by two lines of pipe. Comparison of readings at the meter with the recorder shows no error due to the long connecting pipes. The curves of Fig. 2 are used in conjunction with these daily meter charts to obtain the average rate of flow for each day, which with the calorific value, the kilowatt output of the generators and the generator efficiency, gives the data required for computing the daily average thermal efficiency at the engine shaft. The chart shown in Fig. 36 of the paper represents a flow of 22.044 cu. ft. per min. Other observations and computations for that day were as follows:

B.t.u. per cu. ft. at 62 deg. fahr.	87.1
Average load, kilowatts	7306
B.t.u. per kw-hr	15761
B.t.u. per b.h.p-hr. at 96 % generator efficiency	11311
Thermal efficiency at shaft	22.5

The monthly averages of these daily data, and the results for the year 1909 are shown in Fig. 38 of the paper.

7 Questions are often raised regarding the amount of dust deposited in meters for blast furnace gas. The 60-in. meter has been examined at six-month intervals. The first inspection showed a slight accumulation of moist dust at and near the throat, but not in sufficient quantity to affect the results appreciably; at the second examination no dirt was found; at the third a considerable coating of dirt was found and was cleaned out, unfortunately without accurate measurement of the average thickness. As a means of determining the effect of the reducing diameter on the flow of gas, a comparison was made of the average thermal efficiency for a week preceding and a week following the cleaning as follows:

Week preceding cleaning..16.9 per cent
Week following cleaning..19.0 per cent
Reduction of flow...11 per cent

As far as known no change occurred at the engines to affect the efficiency; consequently, since the flow through the meter varies directly as the area, the reduced diameter due to dust was $\left(\frac{16.9}{19.0}\right)^{\frac{1}{2}}$ x 20 in. = 18.8 in. Therefore the average thickness of the coating was approximately 0.6 in.

8 The cause of this unusual deposit is ascribed to one of the furnaces making special irons, ferrosilicon and spiegel, during the latter part of August and the entire months of September and October 1909, or about 68 days during which the amount of dust found in the raw gas was excessive, as explained in par.73 of the paper. On this basis it is assumed that the deposit began late in August and continued at uniform rate through October, when the error amounted to a maximum of 11%. Suitable corrections were made on the monthly averages shown in the tables and charts. To prevent a repetition of the accumulations of dust in the venturi meter a system of spray nozzles (shown in Fig. 23*b* of the paper) was installed, for flushing the meter throat thoroughly with high-pressure water.

A COMPARISON OF LATHE HEADSTOCK CHARACTERISTICS

By Prof. Walter Rautenstrauch, New York

Member of the Society

The discovery of the properties of high-speed steels, and the large amount of experimental data available on the performance of these steels on various classes of materials, have urged the designer to attempt to incorporate in machine tools such characteristics as will adapt the machines to the most efficient use of the new steels. There exist at present many machines which are intended to meet the new standard of performance, and it will be interesting to examine the results of the attempts which have been made to meet the new conditions and to note the direction in which they have tended. There are many bases on which machine tools may be compared, and no single machine will ever prove best from all points of view; as the limits of this paper prevent the discussion of all these points, one of several possible standards will be adopted as a basis for comparison, and the results will be interesting though not conclusive.

2 Since the new steels will take heavier cuts than is possible with the carbon steels, and still retain their durability, a standard of comparison will be established on the basis of those characteristics of speed and torque in a lathe headstock which permit the most economic removal of shavings from a given class of material, viz., soft and medium steels. A comparison of the speeds and torques actually obtainable in any machine with the standard characteristics will serve as a means for judging the efficiency of the headstock in this particular. In this connection the method devised by Dr. J. T. Nicolson, of Manchester, is employed, the foundations of which are as follows:

3 Since the volume of metal removed by a lathe tool in a given time is a product of the area of cut and the speed of cutting, the weight removed in one minute will be equal to the area of cut in square inches times the speed of cutting in inches per minute times the weight of

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the metal per cubic inch. The force on the tool has been determined experimentally[1] to be approximately proportional to the area of the cut, the torque required to take any size cut is equal to the force on the tool times the radius of the work, and the speed at which the cut can be taken on any diameter of work depends on the spindle speed which can be obtained. These facts, together with the relations which have been established between possible maximum cutting speed and area of cut on different materials,[2] show that in any machine a definite relation must exist between the spindle speeds and the accompanying torques obtainable, that the machine may be adaptable to efficient weight removal on all diameters of any material.

4 The results of the experiments made by the Manchester Association of Engineers and the Berlin Section of the Verein Deutscher Ingenieure, have been used by Dr. Nicolson to derive equations expressing the approximate relation between the area of cut and the maximum cutting speed. The duration of cut was not less than 20 minutes, without injury to the tool. The following result was obtained for the materials in question (medium and soft steel):

$$V = \frac{1}{a} + 15 \qquad [1]$$

where V = cutting speed in feet per minute.
a = area of cut in square inches.

This equation, therefore, serves to determine the cutting speed at which it is possible to operate on this material without injury to the tool, when taking a cut of a given size.[3]

5 To establish a basis for determining the spindle speeds and torques required to remove the maximum weight of shavings on all diameters of work, it is necessary to determine the average area of cut which a lathe of given size should be expected to take. This was accomplished by Dr. Nicolson through correspondence with lathe builders, and the conclusion reached[4] was that the following rule met with wide acceptance for the machining of mild steel forgings:

$$a = \frac{S^2}{25{,}600} \qquad [2]$$

where a = area of cut in square inches.
S = swing of lathe in inches.

[1] Transactions, vol. 25, p. 656.
[2] Report of Manchester Association of Engineers, October 24, 1903.
[3] The Engineer (London), April 7, 1905.
[4] The Engineer (London), April 28, 1905.

6 If the above relations are true, namely, that the maximum possible cutting speed for mild steel varies with the area of cut as expressed in Equation 1; that the average area of cut on this material which a lathe of any given swing should be expected to accommodate is as given in Equation 2; and that the force in the tool varies directly as the area of the cut (for mild steel the force on the tool is approximately 100 tons for each square inch of area cut): then the following basis may be established for the design of, say, an 18-in. lathe capable of removing the maximum weight of shavings in a given time on all diameters of work.

standard area of cut on all diameters

$$= \frac{S^2}{25,600} = \frac{18^2}{25,600} = 0.0126 \text{ sq. in.}$$

$$\text{Force on tool} = 100 \times 2000 \times 0.0126 = 2520 \text{ lb.}$$

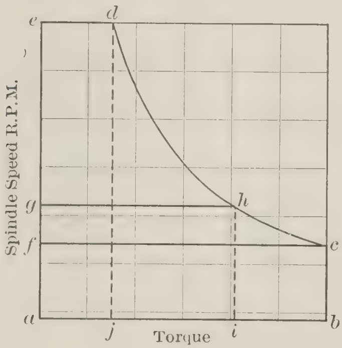

FIG. 1 IDEAL SPEED-TORQUE DIAGRAM

$$\text{torque on work of face plate diameter} = 2520 \times \frac{3}{4} = 1890 \text{ ft. lb.}$$

$$\text{maximum cutting speed at which cut may be taken} = \frac{1}{a} + 15$$

$$= \frac{1}{0.0126} + 15 = 94\tfrac{1}{2} \text{ ft. per min.}$$

revolution of spindle required for this cutting speed on work of face

$$\text{plate diameter} = \frac{94\frac{1}{2}}{\frac{3}{4} \times 2\,\pi} = 20 \text{ r.p.m.}$$

7 The maximum torque required of the lathe will on this basis be equal to 1890 ft. lb., while the minimum spindle speed necessary to give the maximum cutting speed on this area of cut at face plate diameter is 20 r.p.m. When the standard area of cut is taken on a smaller diameter the resulting torque will obviously be less. It will be

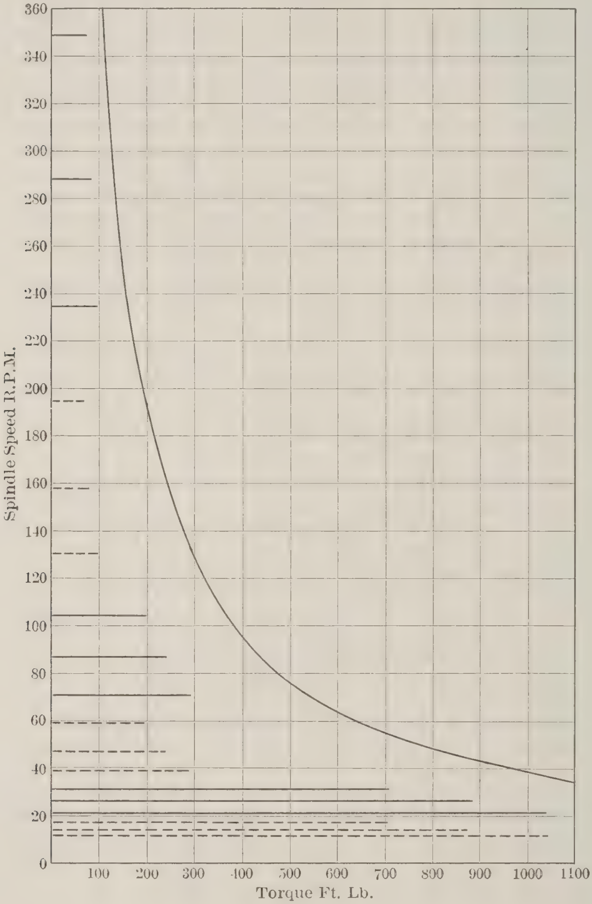

FIG. 2 ACTUAL SPEED-TORQUE DIAGRAM

necessary, however, to increase the spindle speed of the machine, if that surface speed is to be maintained, which is desirable for the most efficient use of the tool and to maintain constancy of volume removed.

8 Since the spindle speed for these conditions will vary inversely as the diameter of the work, and the torque directly as the diameter

of work, it will be obvious that when the problem is to remove the maximum weight of shavings on all diameters of work, the product of speed and torque should be a constant. The highest spindle speed for which the lathe should be designed will depend on the smallest diameter of work which the lathe can economically handle, and the maximum cutting speed desirable on this diameter. On the basis that the least diameter is $\frac{S}{16}$, and that a cutting speed of 120 ft. per min. should be provided, the maximum spindle speed should be

$$Ng = \frac{12 \times 120}{\frac{S}{16}\pi} = \frac{7200}{S} \text{ (approx.)}$$

TABLE 1 (FIG. 2) 18-IN. LATHE

COUNTERSHAFT SPEEDS 195 AND 235 R.P.M.; CONES 13 IN., $10\frac{7}{8}$ IN., $8\frac{3}{4}$ IN. DIAMETER; FIRST BACK-GEAR RATIO 3.31 to 1; SECOND BACK-GEAR RATIO 10.95 to 1, BELT $3\frac{1}{2}$ IN.; ASSUMED BELT PULL 50 LB. PER INCH OF WIDTH

Spindle Speed r.p.m.	Torque ft. lb.	Spindle Speed r.p.m.	Torque ft.lb.
12.00	1040	71.00	314
14.40	865	87.30	261
17.80	700	105.40	212
21.46	1040	131.25	95
26.40	865	158.17	79
31.87	700	195.00	64
39.65	314	235.00	95
47.80	261	289.00	79
58.90	212	349.00	64

For the case of an 18-in. lathe this would result in a maximum spindle speed of 400 r.p.m.

9 In accordance with the above analysis, the ideal characteristic to which the design should tend is as shown in Fig. 1. The abscissæ represent torques, and the ordinates revolutions of the spindle. For the 18-in. lathe the dimensions of the diagram shown in Fig. 1 are

$ab = fc = 1890$ ft. lb.
$af = bc = 20$ r.p.m.
$ae = jd = 400$ r.p.m.
$aj = ed = 95$ ft. lb.

10 Since the product of speed and torque should be a constant, for the reasons previously explained, an equilateral hyperbola be-

tween the points *d* and *c* completes the construction of the ideal diagram. Accordingly, *af* is then the speed at which the spindle should run that the standard area of cut may be taken at its proper speed on work of face-plate diameter, and *fc* is the corresponding torque permitting this area of cut to be taken. Likewise, if the diameter of work is less than face-plate diameter, and since the torque varies directly as the diameter of work for a given area of cut, the torque for diameter of work equal to $S\frac{ai}{ab}$ and standard area of cut, is *gh*, while the spindle speed required to give the appropriate cutting speed is *ag*.

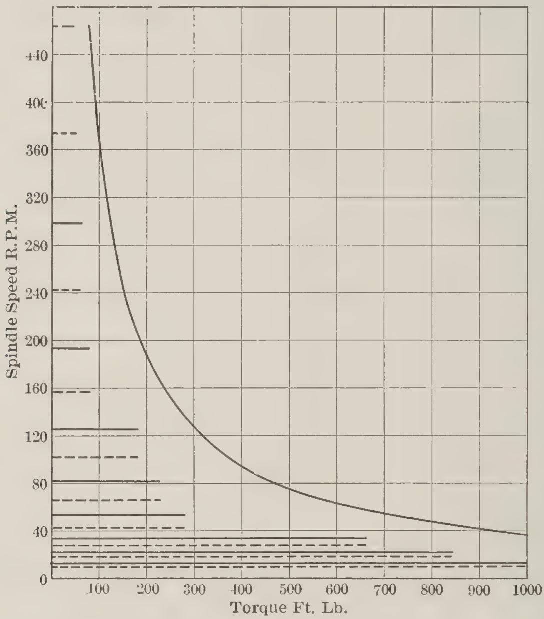

FIG. 3 ACTUAL SPEED-TORQUE DIAGRAM

11 It will be apparent that this diagram may be used in two ways; first, as a means for determining the proper relations which should exist between spindle speeds and torques when a lathe of any size is to be designed for conditions as defined above (to remove the maximum weight of shavings in a given time on all diameters of work of any given material); second, as a means for determining the extent to which the speeds and torques of a lathe already designed correspond to the standard established. In this latter connection it will also be

possible to determine whether or not any speeds, with their corresponding torques, might be omitted without hindering the weight-removing capacity of the headstock.

12 To illustrate the manner in which the diagram may be used as a standard for comparison, Figs. 2 and 3 are presented. In these figures are shown the speeds and torques obtainable in two lathes of recent manufacture, made by different firms. The data for the determination of the speeds and torques, which were obtained from the manufacturers' catalogues, are given in Tables 1 and 2.

13 On the basis that these lathes should be capable of operating on mild steel, with an area of cut on all diameters as determined in the above analysis and up to the maximum cutting speed which the

TABLE 2 (FIG. 3) 18-IN. LATHE

Countershaft Speeds 196 to 234 r.p.m.; Cones 12 in., $9\frac{7}{8}$ in., $7\frac{3}{4}$ in. Diameter; First Back Gear, Ratio 3.66 to 1; Second Back Gear, Ratio 13.5 to 1; Belt 3 in.; Assumed Belt Pull 50 lb. per Inch of Width

Spindle Speed r.p.m.	Torque ft.lb.	Spindle Speed r.p.m.	Torque ft.lb.
11.6	1012	82.0	226
14.4	1012	102.0	177
18.0	835	126.0	177
22.3	835	157.0	75
27.8	655	195.0	75
34.5	655	243.0	61.6
42.7	274	300.0	61.6
53.0	274	375.0	48.5
66.0	226	465.0	48.5

durability of the tool steel will permit, it will be noted that these designs are deficient; for example, if it were required to turn a piece of mild steel 9 in. in diameter, with a cut of 0.0126 sq. in. = $\frac{1}{8}$ in. x $\frac{3}{32}$ in. (approximately) the torque required would be

$$0.0126 \times 2{,}000{,}000 \times \frac{9}{2 \times 12} = 985 \text{ ft. lb.}$$

The lathe illustrated in Fig. 2 would have to take this on spindle speed 1 or 4. Speed 4 would give the highest cutting speed which would be

$$21.46 \times \frac{9 \times \pi}{12} = 50\tfrac{1}{2} \text{ ft. per min.}$$

But with the above area of cut a cutting speed of $94\frac{1}{2}$ ft. per min. would be possible under ideal conditions, hence the minimum time in

which one pound of shavings could be removed under the actual circumstances is about twice what it would be if the required torque were available at the maximum cutting speed.

14 Any number of examples could thus be worked out to illustrate the limits which the dimensions of this headstock impose on either the area or speed of the cut which can be taken on any diameter of work. The question may be asked, to what extent do each of these speeds, with their corresponding torques, contribute to the weight-removing capacity of the lathe, when operating on this material?

15 Referring to Fig. 1, it will be noted that the area $ab \times bc$ is the product of the torque and spindle speed and is

$$f\ a\ r \times \frac{V}{2\ \pi\ r} = \frac{f}{2\ \pi}\ a\ v = K\ a\ V$$

where

f = force on tool in pounds per square inch.
a = area of cut in square inches.
V = cutting speed in feet per minute.
r = radius of work in feet.

But the area of the cut times the speed of cutting is a measure of the volume of metal removed in a given time and hence a measure of the weight removed in a given time. Any condition, therefore, fixing the limits to the area and speed of cut which can be taken on any diameter of work will limit the maximum rate at which metal can be removed.

16 Let us determine, therefore, to what extent the gap between the speeds, and the departure of the torque from that which has been established as desirable at the different speeds, will effect the weight-removing capacity of the lathe. Let Fig. 4 represent the ideal torque-speed diagram for any lathe, established on the above basis, and ab and ac two spindle speeds actually obtainable with torques bd and ec respectively. Then with a speed of spindle ab and torque bd, the standard area of cut may be taken on work of diameter $S\,\frac{bd}{ag}$, where S is the swing of the lathe.

17 Suppose it is only necessary to take a lighter cut $\left(a = \frac{bj}{bd}\right)$ on the same diameter of work, can it be more economically removed by taking the full cut $\left(a = \frac{bj}{bd}\right)$ with the spindle speed ab or, neglect-

ing the time for resetting the tool, to use a still lighter cut with the spindle speed *ac* and go over the work twice to bring it to finished size? Also, up to what limit of area of cut will it be more economical to use the speed *ab* than *ac*?

18 Now the whole area of cut may be taken at the lower speed *ab*, for which the rate of weight removed is represented by $(bj \times jq)$, or it may be taken at the higher speed *ac* by going over the work twice, first with a depth of cut and feed, giving an area of cut equal to $a_s \frac{bx}{bd}$, and again with remaining depth of cut and a feed giving the same area of cut $a_s \frac{bx}{bd}$, required to bring the piece down to size.

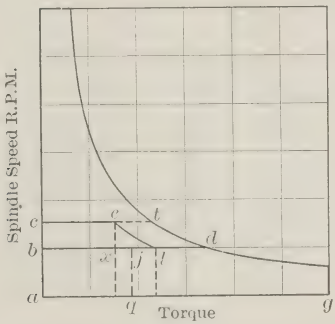

FIG. 4 IDEAL SPEED-TORQUE DIAGRAM

Neglecting for the present the time taken to reset the tool, it will be seen that the weight removed in the same time will be greater by going over the piece twice, each time with an area of cut equal to $a_s \frac{bx}{bd}$, and spindle speed *ac*, than by taking the cut of area $a_s \frac{bj}{bd}$ at the lower spindle speed *ab*.

19 To illustrate more specifically, suppose for example that *bd* represents the torque required to take the standard cut a_s on some given diameter of work, then *bj* would represent the torque required on the same diameter of work when the area of cut is not equal to the standard area but is equal to $a_s \frac{bj}{bd}$, since the ratio of the torques is equal to the ratios of the areas of cut on the same diameter of work.

If a_s = 0.0126 sq. in., $a_s \frac{bj}{bd}$ = 0.0084 sq. in., $a_s \frac{bx}{bd}$ = 0.0063 sq. in., and ab = 30 r.p.m., ac = 50 r.p.m., then the rate of weight removal when taking the area of cut $a_s \frac{bj}{bd}$ at the spindle speed ab is proportional to 0.0084 x 30 = 0.252, while if the area of cut $a_s \frac{bx}{bd}$ is taken at the spindle speed ac the rate of weight removal is proportional to 0.0063 x 50 = 0.315. The above condition will be true up to such areas of cut on the given diameter which, when multiplied by the lower spindle speed, will give a rate of weight removal greater than 0.315. This limit of area of cut may be conveniently determined by drawing an equilateral hyperbola through e and letting it cut bd at l.

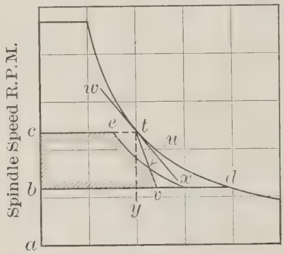

Fig. 5 Ideal Speed-Torque Diagram

Areas of cut from $a_s \frac{bl}{bd}$ to a_s can be more economically removed for the given diameter of work at the lower speeds, while areas of less than $a_s \frac{bl}{bd}$ can be more economically removed by taking the lighter areas of cut $a_s \frac{bx}{bd}$ at the higher spindle speed and going over the work twice to bring it to size. Therefore, the efficiency in weight removed of this range of speeds and torques, compared to the ideal case where all speeds and torques, define within the area $bctd$ are available, is represented by the ratio

$$\frac{\text{area } bcel}{\text{area } bctd}$$

In case the areas of cut from $a_s \frac{bx}{bd}$ to $a_s \frac{bl}{bd}$ can not be taken on the diameter in question at the higher spindle speed because the resulting surface speed is too great, the above statement is not true. A few of the designs examined have been checked in this manner and found to come within the limit just defined.

20 No allowance, however, has been made for the time required to run the carriage back and reset the tool. It will be seen that as the limiting area $a_s \frac{bl}{bd}$ is approached, the time saved on the use of the lower speed in place of the higher becomes less. Accounting for the time required to reset the tool for a second run, it will be noted that the limiting area is reached before $a_s \frac{bl}{bd}$. Just where the limit will be encountered it is impossible to determine except by empirical methods. Dr. Nicolson has ascertained that this limit may be approximately determined by the use of the following construction, irrespective of the type or design of the lathe.

21 Let Fig. 5 represent the conditions taken in Fig. 4. Construct a tangent *wx* to the hyperbola at *t* and drop the vertical *ty*. Bisect the angle between *wx* and *ty* by the line *tv*. The efficiency of this particular part of the headstock will be approximately represented by

$$\frac{\text{area } ceuxb}{\text{area } ctdb}$$

Areas of cut equal to and greater than $a_s \frac{bv}{bd}$ can be more economically taken on this diameter of work at the lower speed *ab* because of the difference in time required to handle the machine for the two cuts required to bring the piece to size.

22 This constructionis to be considered as a rough approximation only, and represents the facts as well as the conditions in the case will permit. This method of comparing the efficiency of a lathe with a predetermined ideal performance on any given material is due to Dr. J. T. Nicolson and Mr. Dempster Smith, to whom all credit should be given. The above method is useful in determining the adaptability of a lathe to meet only one of the many kinds of service in which the lathe may be employed and is not a final means for either justifying or condemning a lathe for general purpose work.

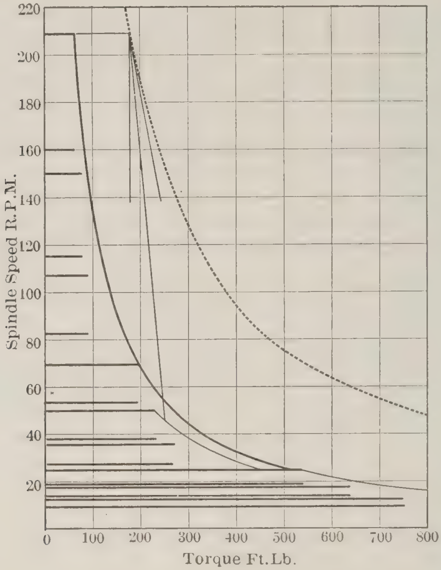

FIG. 6 16-IN. DOUBLE BACK-GEARED LATHE

TABLE 3 (FIG. 6) 16-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS $8\frac{1}{4}$ IN., $9\frac{7}{8}$ IN., $11\frac{1}{2}$ IN.; BELT WIDTH $3\frac{3}{4}$ IN.; COUNTERSHAFT SPEEDS 115 AND 150 R.P.M. FIRST BACK-GEAR RATIO 3 TO 1; SECOND BACK-GEAR RATIO $8\frac{1}{3}$ TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
9.87	748	50.00	231
12.85	748	53.50	193
13.80	642	69.60	193
18.00	642	82.50	90
19.20	537	107.00	90
25.00	537	115.00	77
27.50	270	150.00	77
35.73	270	160.00	65
38.33	231	209.00	65

The torques were computed on the basis of 50 lb. per inch of belt effective on the pulley surface. As a basis of the foregoing analysis, the lathe should be capable of the following:

Ng (greatest desirable spindle speed) = 450 r. p. m.
Ml (least desirable spindle speed) = $28\frac{3}{4}$ r. p. m.
Moximum desirable torque = 1366 ft. lb.

23 There is, however, one point of broad application which a speed torque diagram constructed according to the above basis will immediately bring out; that is, the uselessness of certain speeds, with their corresponding torques, possible in a given lathe on any class of work. As an illustration of how the relative merits of lathes of different make may be determined with reference to a common standard, 11 lathes selected from the catalogues of different builders have been used in the

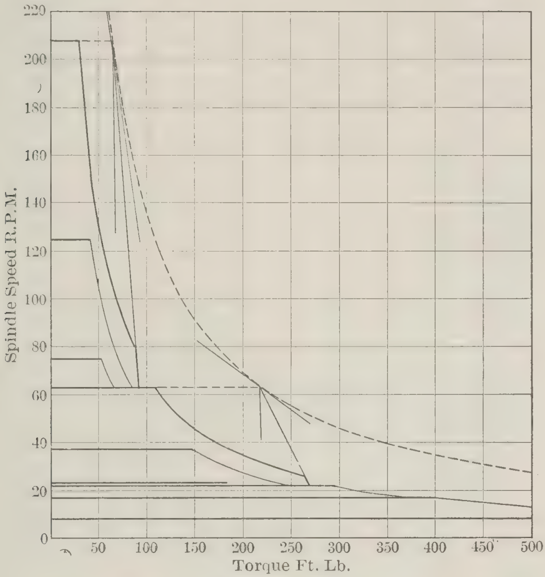

FIG. 7 16-IN. DOUBLE BACK-GEARED LATHE

TABLE 4 (FIG. 7) 16-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS 6 IN., 8 IN., 10 IN.; BELT WIDTH $2\frac{1}{2}$ IN.; COUNTERSHAFT SPEED 125 R.P.M.; FIRST BACK-GEAR RATIO $3\frac{1}{3}$ TO 1; SECOND BACK-GEAR RATIO $9\frac{1}{2}$ TO 1

Spindle Speeds r.p.m.	Torque ft.lb.
7.9	495
17.04	400
21.9	294
22.5	182
37.5	147
63.00	109
75.00	52
125.00	42
208.3	31

Ng = 450 r.p.m.; Nl = $28\frac{3}{4}$ r.p.m.; maximum torque = 1366 ft. lb.

construction of the following figures. In each case the data were obtained from the catalogues, or by correspondence with builders, and the possible speeds and torques determined. The data and results thus obtained are shown in Tables 3 to 13, the corresponding speed-torque diagrams being represented by Figs. 6 to 16.

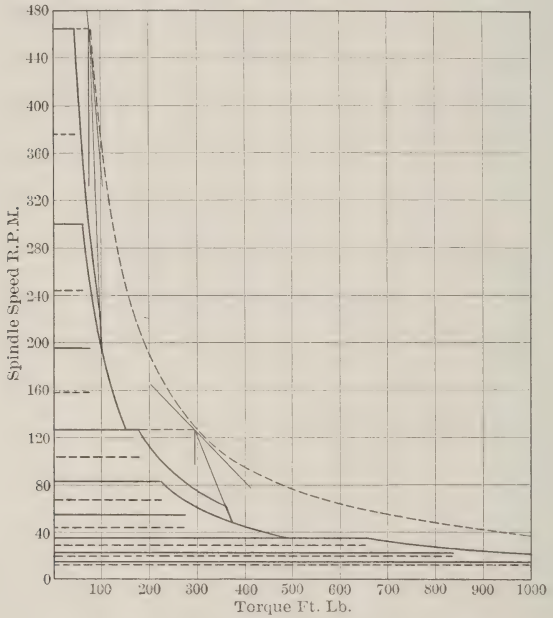

FIG. 8 18-IN. DOUBLE BACK-GEARED LATHE

TABLE 5 (FIG. 8) 18-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS $7\frac{3}{4}$ IN., $9\frac{7}{8}$ IN., 12 IN.; BELT WIDTH 3 IN.; COUNTERSHAFT SPEEDS 196 AND 243 R.P.M.; FIRST BACK-GEAR RATIO 3.66 TO 1; SECOND BACK-GEAR RATIO 13.5 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
11.6	1012	82.0	226
14.4	1012	102.0	177
18.0	835	126.0	177
22.3	835	157.0	75
27.8	655	195.0	75
34.5	655	243.0	61.6
42.7	274	300.0	61.6
53.0	274	375.0	48.5
66.0	226	465.0	48.5

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft. lb.

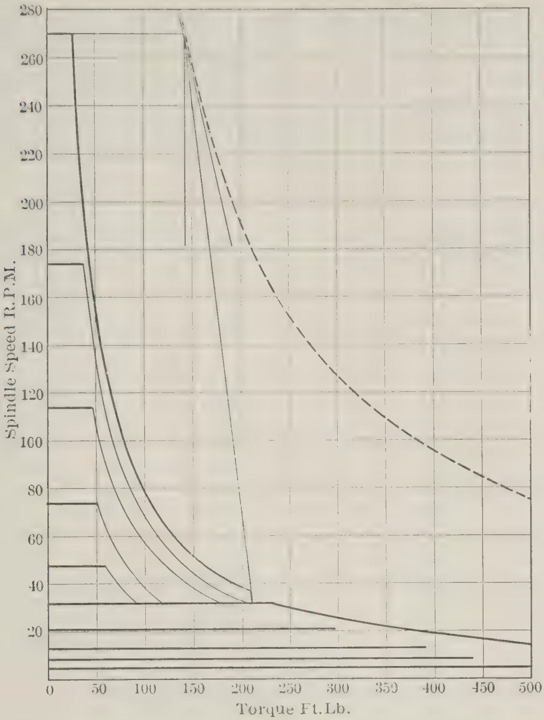

FIG. 9 18-IN. SINGLE BACK-GEARED LATHE

TABLE 6 (FIG. 9) 18-IN. SINGLE BACK-GEARED LATHE

CONE DIAMETERS $5\frac{1}{4}$ IN.; $6\frac{13}{16}$ IN., $8\frac{7}{8}$ IN., $9\frac{15}{16}$ IN., $11\frac{1}{2}$ IN.; BELT WIDTH $2\frac{1}{2}$ IN.; COUNTERSHAFT SPEED 125 R.P.M.; BACK-GEAR RATIO 8.44 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
5.70	500
8.75	437
13.50	390
20.60	300
32.00	231
48.00	60
74.00	52
114.00	46
174.00	36
270.00	27

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft.lb.

24 Among the facts brought out by this method of comparison of the adaptability of different makes of lathes to the performance of a standard task, there are two which are particularly striking. It will be noted in the first place that a considerable difference of opinion exists among the several builders, the characteristics of whose lathes

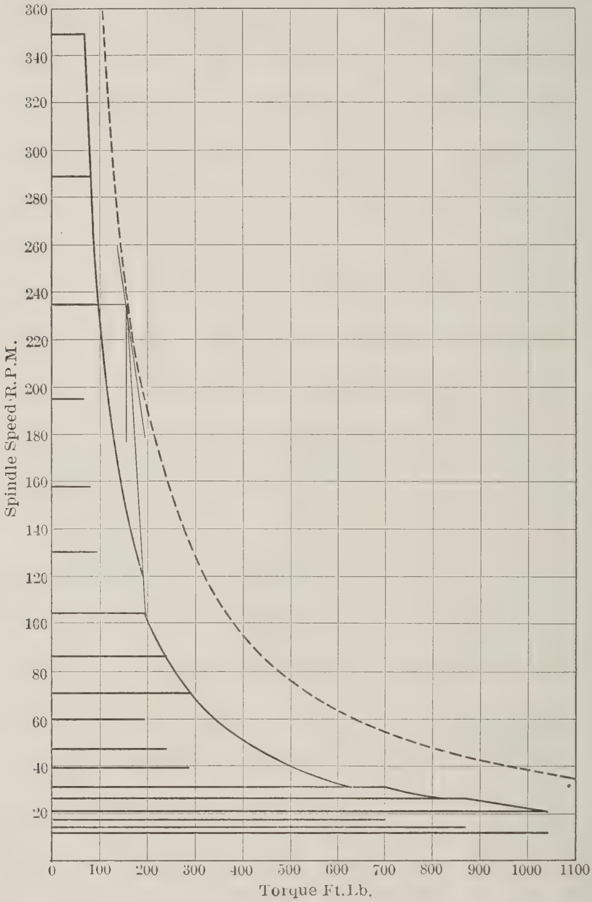

FIG. 10 18-IN. DOUBLE BACK-GEARED LATHE

are here illustrated, as to what constitutes a sufficient powering of the lathe to meet the demands of the high-speed steels, the number of speeds to be furnished, and the manner in which the speeds and torques should be spaced. If in reply to the questions of powering it is stated that the particular lathe in question is intended for taking

lighter cuts, which might be a proper basis for design under certain circumstances, it still remains to justify the manner in which the speeds and torques are spaced.

25 For example, take the case of the lathe represented in Fig. 6. For the single instance of having to turn a 9-in. piece of soft steel it

TABLE 7 (FIG. 10) 18-IN. DOUBLE BACK-GEARED LATHE

CONE DIAMETERS $8\frac{3}{4}$ IN., $10\frac{7}{8}$ IN., 13 IN.; BELT WIDTH $3\frac{1}{2}$ IN.; COUNTERSHAFT SPEEDS 195 AND 235 R.P.M. FIRST BACK-GEAR RATIO 3.31 TO 1; SECOND BACK-GEAR RATIO 10.95 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
12.0	1040	71.00	314
14.4	865	87.3	261
17.8	700	105.4	212
21.46	1040	131.25	95
26.4	865	158.17	79
31.87	700	195.00	64
39.65	314	235.00	95
47.8	261	289.00	79
58.9	212	349.00	64

Ng = 400 r.p.m.; Nl = 20 r.p.m.; maximum torque = 1900 ft. lb.

TABLE 8 (FIG. 11) 20-IN. ROUGHING LATHE

CONE DIAMETERS $11\frac{1}{2}$ IN. AND 13 IN.; 6-IN. DOUBLE BELT; COUNTERSHAFT SPEEDS 340 AND 365 R.P.M.; FIRST BACK-GEAR RATIO 3 TO 1; SECOND BACK-GEAR RATIO 6 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
50	1360	128	680
53	1210	137	605
64	1360	300	227
68	1210	323	202
100	680	384	227
107	605	412	202

Ng = 360 r.p.m.; Ml = 16 r.p.m.; maximum torque = 2666 ft. lb. Double belts are estimated as having 75 lb. per inch of width effective on pulley surface.

will be seen that the maximum area of cut that can be taken is limited to 0.01 sq. in., equivalent to a cut $\frac{1}{6}$ in. by $\frac{1}{16}$ in., and that the highest speed which the resulting torque of 750 ft. lb. will permit is 12.85 r.p.m., giving a cutting speed of 30 ft. per min. on this diameter. The cutting speed possible with soft steel on this area of cut is approximately 115 ft. per min. or if an area of cut of 0.0036 sq. in. is to be

taken on the same diameter, the highest spindle speed which the resulting torque of 270 ft. lb. will permit is 35.73 r.p.m., giving a cutting speed of 85 ft. per min. The cutting speed possible with this area of

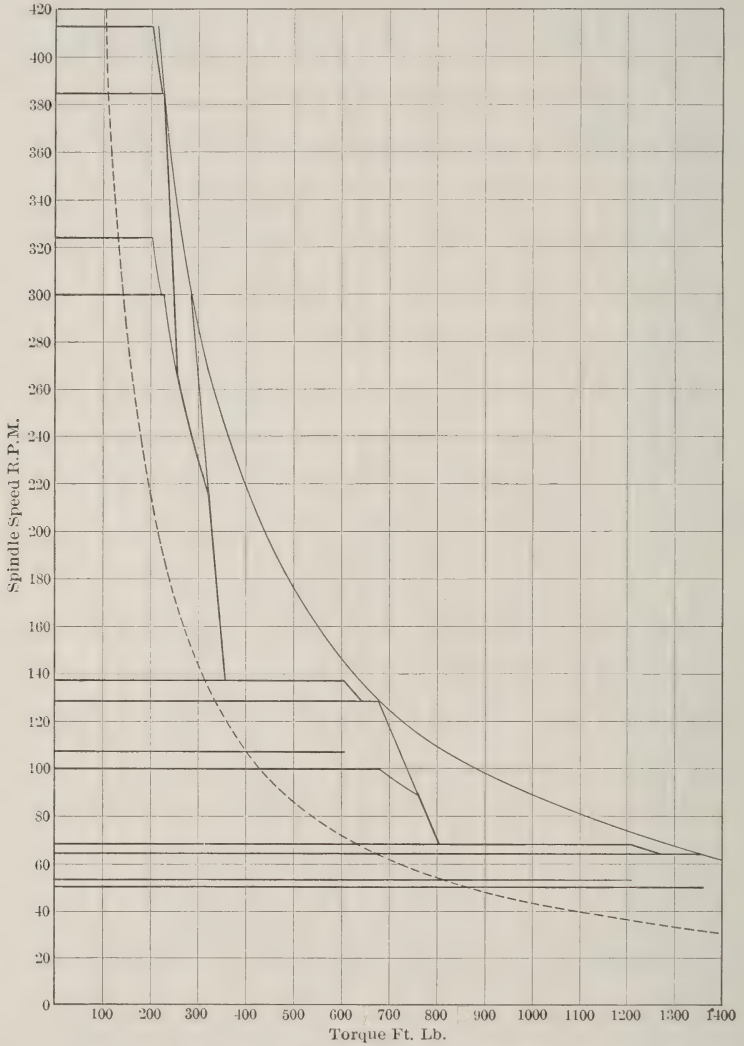

Fig. 11 20-in. Roughing Lathe

cut is above 200 ft. per min. For this size of work, then, the lathe is inefficient, or for efficient operation is limited to forms of work in which the cutting speeds and area of cut determined are the highest

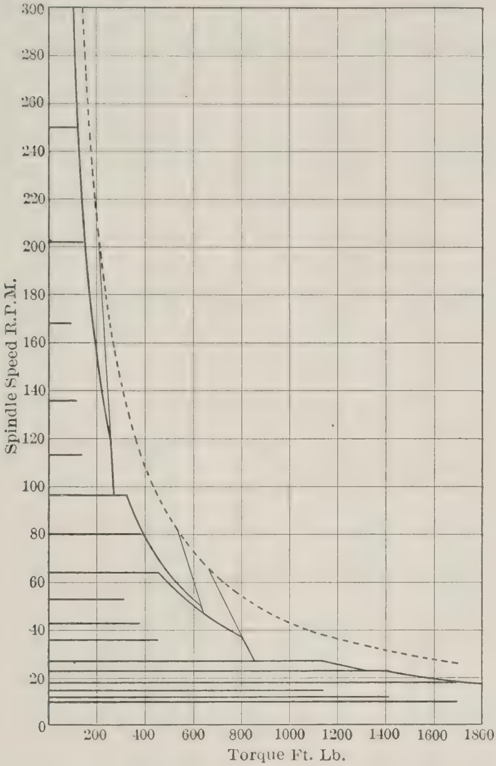

Fig. 12 21-in. Heavy-Duty Lathe

TABLE 9 (FIG. 12) 21-IN. HEAVY-DUTY LATHE

Cone Diameters $10\frac{3}{4}$ in., $13\frac{3}{8}$ in., 16 in.; Belt Width $4\frac{1}{2}$ in.; Two Countershaft Speeds; First Back-Gear Ratio 3.13 to 1; Second Back-Gear Ratio 11.3 to 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
12	1,700	64	470
10	1,420	80	394
15	1,150	96	320
18	1,700	113	150
23	1,420	136	126
27	1,150	168	102
36	470	202	150
43	394	250	126
53	320	300	102

$Ng = 342$ r.p.m.; $Nl = 14$ r.p.m.; maximum torque = 3087 ft. lb.

possible. In like manner, the limits of performance on any other diameter of work imposed by the torque-speed characteristics of the lathe, may be determined.

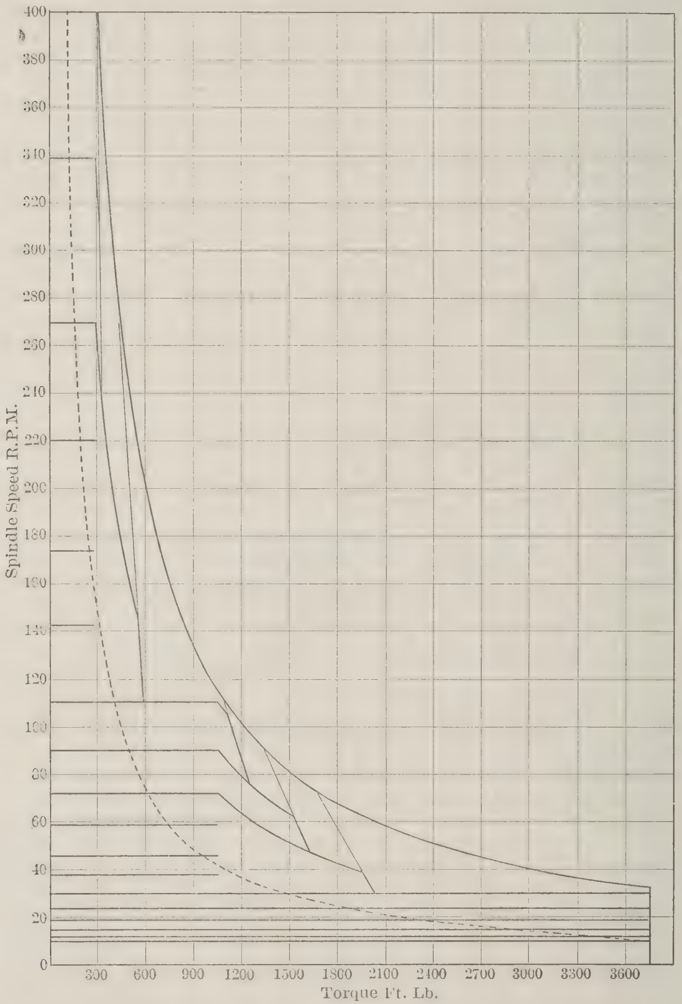

FIG. 13 24-IN. GEARED-HEAD LATHE

26 The extent to which the several speed-torque characteristics supplement one another is also very conveniently brought out in these diagrams. Again referring to Fig. 6, it will be noted that the con-

tribution of a number of the speed-torque combinations to the efficiency of the lathe for weight removal is brought into question, no matter what the standard of performance may be. If the foregoing analysis is rational, it indicates that the speeds 160, 150, 115, 107, 82, 73, 53, 50, 38, 35, 27, 19, 20, 18, 13.8, 12.85, and 9.87, with their accompanying torques are superfluous. Only upon a sufficient increase in the

TABLE 10 (FIG. 13) 24-IN. GEARED-HEAD LATHE

COUNTERSHAFT PULLEY 16 IN.; HEADSTOCK PULLEY $15\frac{15}{16}$ IN.; COUNTERSHAFT SPEEDS 205 AND 250 R.P.M.; $6\frac{1}{2}$-IN. DOUBLE BELT; FIRST BACK-GEAR RATIO 3.69 TO 1; SECOND BACK-GEAR RATIO 13 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.	Spindle Speeds r.p.m.	Torques ft.lb.
10	3760	72	1066
12	3760	90	1066
15	3760	110	1066
19	3760	143	289
24	3760	174	289
30	3760	220	289
38	1066	270	289
46	1066	339	289
59	1066	414	289

Ng = 300 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

TABLE 11 (FIG. 14) 24-IN. GEARED-HEAD LATHE

COUNTERSHAFT PULLEY 16 IN.; HEADSTOCK PULLEY 16 IN.; BELT WIDTH 5 IN.; COUNTERSHAFT SPEED 400 R.P.M.; BACK-GEAR RATIO 5 TO 1

Spindle Speeds r.p.m.	Torques ft.lb.
21	3174
32	2083
40	1666
60	1111
107	623
160	417
200	334
300	223

Ng = 300 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

corresponding torque can each of these speeds add to the efficiency of the lathe. Considered on the basis of a dead investment alone, it will be seen that the equipment required to give the above speeds, which seem without justification, adds a useless burden to the product of this machine.

27 An examination of some of the following diagrams will reveal facts similar to those announced above. In those cases where two countershaft speeds are employed it will be noted that no increase in efficiency is had from this source. It is a pleasure, however, to note some exceptions, particularly in the case of Fig. 14. It will be observed that upon this basis of analysis there is a justification for each speed-torque characteristic. If any of the speeds were cut out, the efficiency

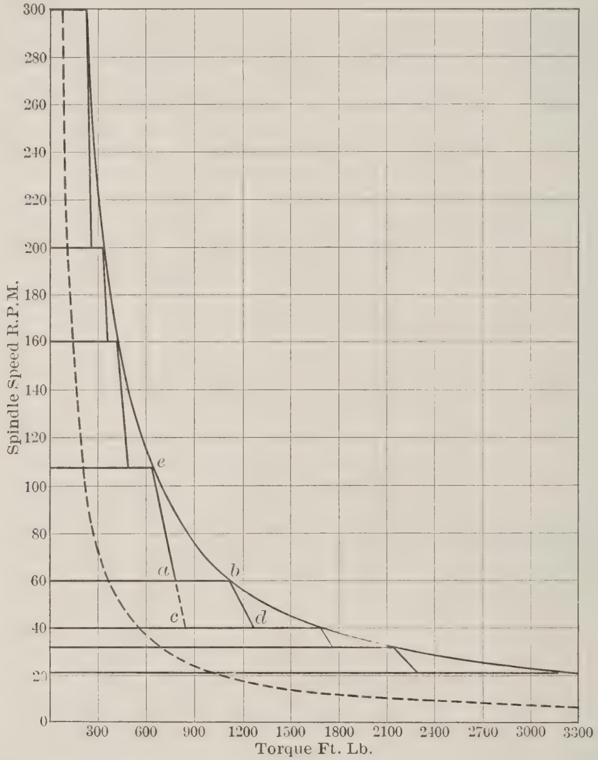

Fig. 14 24-in. Geared-Head Lathe

of the lathe would be reduced. With respect to the standard task used in this discussion, it will be noted that the removal of speed 60 from the headstock would reduce the efficiency by an amount proportional to the area *abcd*.

28 Another matter which appears in this connection is the relation of the efficiency to the number of speed-torque combinations, of which only eight are possible in this lathe. To what extent would the effi-

ciency be increased if eight additional speeds, with their accompanying torques, should be spaced halfway between the present combinations? The answer to this question would be obtained by the same method

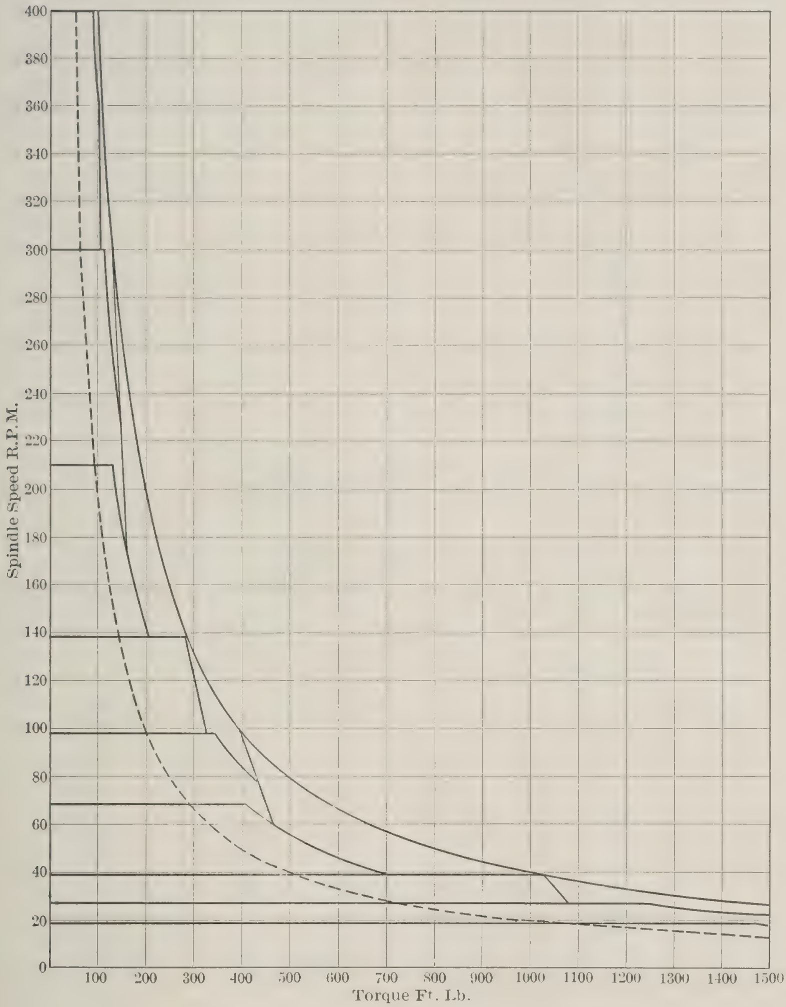

FIG. 15. 24-IN. DOUBLE BACK-GEARED LATHE

by which it was determined that the omission of speed 60 in the previous problem would reduce the efficiency proportional to the area *abcd*.

29 An analysis of this sort will show two things: first, that increasing the number of speeds without regard to the torque does not necessarily increase its adaptability to economic performance; second, that the amount by which the efficiency can be increased does not increase in direct proportion to the additional amount of speed changes provided,

TABLE 12 (FIG. 15) 24-IN. DOUBLE BACK-GEARED LATHE

Cone Diameters 10½ in., 12¾ in., 15 in.; Belt Width 4½ in.; Countershaft Speed = 300 r.p.m. First Back-Gear Ratio 3.1 to 1; Second Back-Gear Ratio 11.1 to 1

Spindle Speeds r.p.m.	Torques ft.lb.
10	1476
27	1254
39	1032
68	410
98	350
139	288
210	133
300	113
429	93

Ng = 380 r.p.m.; Nl = 9.9 r.p.m.; maximum torque = 4608 ft. lb.

TABLE 13 (FIG. 16) 26-IN. "MASSIVE" LATHE

Cone Diameters 7 in., 9¾ in., 12½ in., 15¼ in., 18 in.; Belt Width 4 in.; Countershaft Speeds 125 r.p.m.; Back-Gear Ratio 12 to 1

Spindle Speeds r.p.m.	Torques ft.lb.
4.05	1800
6.65	1512
10.40	1250
16.3	975
26.8	700
48.6	150
80.0	126
125.0	104
196.0	81½
322.0	58¾

Ng = 277 r.p.m.; Nl = 8.15 r.p.m.; maximum torque = 5860 ft. lb.

even if the accompanying torques are properly determined. If 24 speed-torque combinations were properly spaced in the design represented in Fig. 14, the increase in efficiency over the eight already presented would not be twice as much as if 16 speed-torque combinations should be introduced in the same manner.

30 Closely associated with the matter of the increase in efficiency by the introduction of additional speed-torque changes is the problem of whether or not the increase is warranted by the increase in cost due to the additional equipment, and whether the management of the shop is such as to insure proper use of the additional equipment. The latter is in general the more vital question. In fact, the whole matter

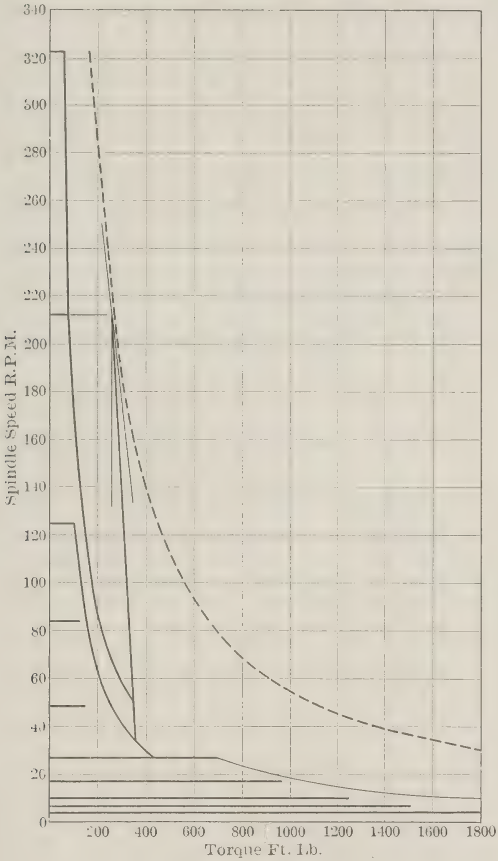

FIG. 16 26-IN. "MASSIVE" LATHE

of the efficiency of any machine as a part of a plant is as largely dependent on the management as upon the design of the machine.

31 But confining our attention particularly to questions of design, we note another field of usefulness for this method of analysis. It affords a means for determining the beneficial effects of motor equipment

on the efficiency of the machine. Given any particular machine with certain possible speed-torque combinations, what changes can be wrought by the use of direct motor drive when the motor has certain characteristics of speeds and torques? The limits of this paper will not permit a full discussion of this question, but it is pointed out as one way of determining the effect of motor drive on efficiency which will lead to more definite conclusions than any number of photographs illustrating the neater appearance of a motor-equipped machine over a belt-driven machine.

32 In conclusion it may be remarked that there seems to be need for a more rational method of procedure in determining the speed-torque characteristics of a lathe. While it is impossible to formulate all the conditions which a lathe may encounter in its operation, at the same time it is believed that a method of analysis such as that described in this paper will materially assist the designer in determining the speed-torque relations which are justifiable, and will enable the purchaser to determine whether or not the speed-torque characteristics of any given lathe are adaptable to his conditions.

FINISHING STAY-BOLTS AND STRAIGHT AND TAPER BOLTS FOR LOCOMOTIVES

By C. K. Lassiter,[1] Richmond, Va.

Non-Member

The locomotive boiler of average size contains about 1500 staybolts, the number varying from 1200 in the smaller sizes to 2000 or more in the heavier types. They vary in length from $4\frac{1}{2}$ in. to $10\frac{1}{2}$ in. for the water-space bolts, which constitute about 75 per cent of the total number, to about 28 in. for the radial and crown bolts.

2 Probably no part of the boiler is subject to more destructive conditions than these little staybolts. The most serious strains are those due to expansion and contraction of the inner sheet, which bend the bolts and cause them to break close to the outer sheet. This is especially true of the side or water-space stays, which are comparatively short and have very little flexibility.

3 The material used is a high grade of refined iron, close-grained and tough. The pitch being very important on account of entering the second sheet, these stays were formerly cut to length from the bar, drilled for centers, and threaded on engine lathes. The center-drilling was not always concentric and considerable time was required to center the rough bolt so that a good thread could be obtained. This method proving too expensive, bolt cutters were used for the work, but the results were not entirely satisfactory. It was difficult to cut the threads full and smooth with one passage of the chasers and the second passage was taken at the sacrifice of pitch, as well as of time, because there was not enough material to remove to carry the chasers along properly. The introduction of the lead screw in bolt cutters brought about a very considerable improvement in pitch, but still there was trouble in getting the thread smooth for the reason

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[1] Mechanical Superintendent, American Locomotive Company.

that the chasers were not always as accurate as the lead screw, under which conditions the threads would be rough or torn.

4 About thirty years ago the idea was conceived of concaving the bolts or reducing them in the center below the root of the thread, the object being to provide flexibility to compensate for the expansion between the inner and the outer sheets. Laboratory tests showed that a bolt reduced in the center would withstand about twice as many vibrations before breaking as one on which the threads were left straight for the full length. For many years it was the accepted practice to reduce a bolt in diameter on engine lathes after it was threaded in the bolt cutter and drilled for centers.

5 In 1900, Alonzo Epright, an engineer in the employ of the Pennsylvania Railroad, designed machines which were fully auto-

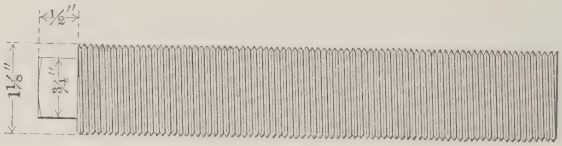

Fig. 1 Square End Water Space Stay (Plain,

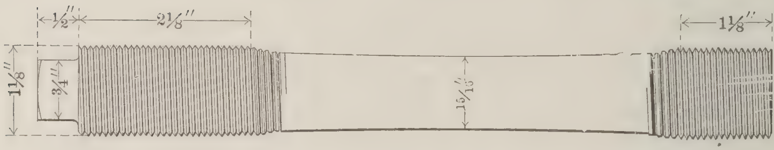

Fig. 2 Square End Water Space Stay (Concave)

matic in that they made from the bar, threaded and concaved, all diameters of side stays up to ten or eleven inches in length. The author has no knowledge of the production of these machines and therefore can make no comparison of costs.

6 The vertical type of machine for threading these bolts was used to some extent and it seemed that if the proper chaser could be made the best results would be obtained from this type of machine because the weight of the head would assist the chaser to give an accurate pitch. In the horizontal or bolt cutter type the chaser must carry along the vise and carriage to the detriment of accuracy in the lead. Also, the flow of oil would assist in washing away the chips, which were troublesome in the horizontal machine. Furthermore,

the vertical type of machine is more convenient to operate, one man attending six or eight spindles with ease.

7 After a great deal of experimenting a die head was developed in which, with chasers properly ground, the limit of accuracy of 0.01 in. in 8 in. can be maintained without the use of the lead screw, which is more nearly a perfect pitch than many staybolt taps in daily use. Where a proper lubricant is used a very fine, smooth thread can be obtained at a uniform cutting speed of 20 ft. per min.

8 The turning or reducing tools are shown in Fig. 3, the cutting points being visible at the center, back of the chasers. To these tools

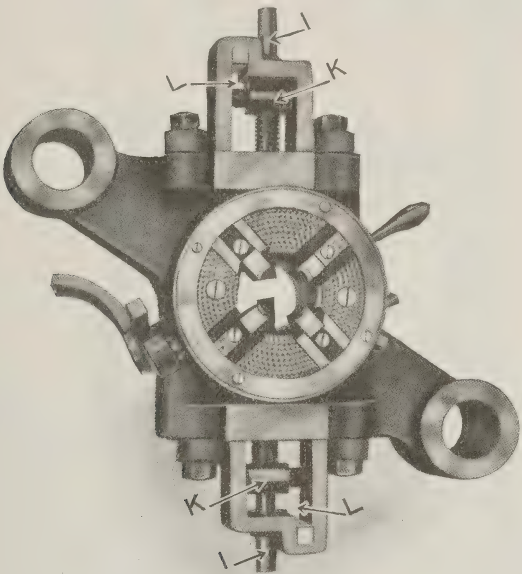

Fig. 3 Die Head for Threading Staybolts

are attached the crossheads *KK*, which are actuated by profilers or formers passing through the spaces *LL*, over which the head is drawn by the chaser, the staybolt acting as a lead screw.

9 The staybolt-threading machine is shown in Fig. 4. The several die heads are attached by small rods to straps passing over the pulleys on a shaft at the top of the machine. The operator grasps one of the strap handles with his right hand and, by the aid of the rotating

pulley over which the strap passes, raises the die head until it comes in contact with the bracket which closes the die. With his left hand he places the squared end of a staybolt in a holder underneath the die and allows the head to drop until the chasers begin to cut, when he moves to the next die head and repeats the operation. By the time he has placed all the heads in operation, the first bolt is finished, the die having dropped automatically when the threading was completed.

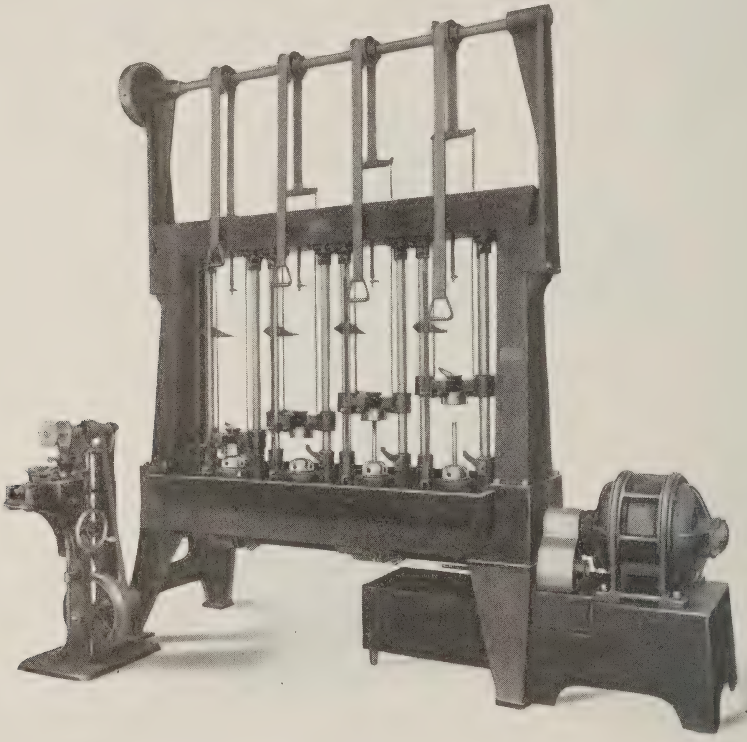

FIG. 4 STAYBOLT-THREADING AND REDUCING MACHINE, WITH SPECIAL GRINDER FOR CUTTING TOOLS

10 In Fig. 4, the die head at the right is shown raised sufficiently to insert the staybolt in place; the next at the left is just beginning to thread the bolt and the two other die heads are in still lower positions.

11 A comparison of costs by the two methods, taking a $7\frac{1}{2}$-in. side stay as an average length, would be about as follows:

FORMER PRACTICE

Threading-in bolt cutter, usually taking two cuts at 20 cents........	$0.40
Drilling for centers	0.22
Concaving or reducing on engine lathe..............................	0.75
Cost per hundred	$1.37

PRESENT PRACTICE

Present cost, threaded the entire length or threaded and concaved for all sizes and lengths, per hundred..................................	$0.13

Using the average number of stays, a saving of labor cost of $18.60 per boiler is obtained with a minimum of rejected stays.

METHODS OF DRILLING STAYBOLTS

12 The telltale holes which are drilled in the staybolts have been the cause of considerable expense and annoyance. Some railroads drill them after the stays are placed in the boiler, with pneumatic hand drills. Under these conditions there is danger that the hole may not be central. It often happens that the drill runs through into the water space or is broken off in the hole. In either case it is necessary to remove the bolts and put in others. Sometimes the holes are drilled on a vertical drilling machine before being placed in the boiler. Even then the breakage of drills is very large, averaging about sixteen to the boiler, and each broken drill means a staybolt thrown away.

13 An automatic machine has been devised for drilling these holes before the stay is placed in the boiler. They are fed from a hopper and automatically centered in position for the drill. When the hole is bored about one-third of the depth, the drill is withdrawn and the bolt is carried forward in the turret mechanism which holds it to a second and a third drill, completing the hole. Each drill is 0.01 in. smaller than the preceding one, providing for a minimum of friction and a maximum of clearance for chips. The holes are of uniform depth and in the center of the bolt. The average breakage is about three or four drills to the boiler.

COMPARISON OF COSTS

Drilling in the boiler, per hundred (to which should be added the cost of replacements	$0.90
Drilling under drill press, per hundred (to which should be added cost of drills and waste of material and labor)......................	0.45
Drilling in the automatic machine, per hundred (with the minimum number of broken drills and bolts destroyed)..................	0.12

METHODS OF FINISHING STRAIGHT AND TAPERED BOLTS

14 The usual method of finishing straight and tapered bolts for locomotives was to drill for centers, place in engine lathes, face under the head, turn the body taper, turn the part to be threaded straight and to proper size, face down the thread end to length and shape, leaving the center intact, test and file to accuracy, and cut off center point, after which the bolt is ready to be threaded in the bolt cutter and to have the hexagon head changed to any special shape desired.

15 About 1889, S. M. Vauclain, Mem.Am.Soc.M.E., designed and used a turning head in connection with a vertical machine for bolts up to 12 in. long. Under rights obtained from him the Pennsylvania Railroad placed an equipment of this kind in its Altoona shops and that is the only railroad known to the author using other than engine lathe methods in finishing bolts.

16 As a great many straight and tapered bolts used in locomotives are 12 in. to 20 in. in length and even longer, it became necessary to design for this work a turning head which would handle taper bolts up to 18 or 20 in. in length and up to $1\frac{3}{4}$ in. diameter of thread, and straight bolts in any length up to 27 in. and up to $2\frac{1}{2}$ in. diameter. It may be quite possible to go beyond these dimensions should the specifications require. These requirements have been met by a special machine of the vertical, multiple-spindle drill type, with which is used a special cutter head shown in Fig. 5. This head is the real or essential means of producing these bolts, either straight or taper and cylindrically true to the axis, the machine being simply a proper means of driving and feeding the bolt during the turning operation.

17 The cutter head consists of a retaining shell of cast iron, the bore of which must be round and straight; six segments, three of which are rigidly fastened to the shell, the other three having a limited amount of freedom and being fastened in place by a taper key with an adjusting screw located in the center of the radius with a bearing on the shell; and three blades, alternating with three guides, placed between the segments and backed up with taper keys and adjusting screws. The taper keys, in connection with a certain amount of taper on the blades and guides, have sufficient movement to provide for about one-eighth inch adjustment for re-grinding of the blades, or with the same amount on the guides, one-quarter inch in diameter of bolts. It will readily be seen that when an accurately ground plug gage of the size that it is desired to turn the bolt is placed centrally in the head, the blades and guides can be adjusted to their proper position. The three

loose segments are then forced forward by the taper key, clamping the blades and guides rigidly in their proper working position.

18 The economical use of this method of turning bolts, particularly in the railroad shops and locomotive works where taper bolts are largely used, necessitates a change of system. The usual practice, especially on repair work, has been to carry in stock only standard sizes of forgings, though in some cases the more common sizes were placed in stock finished. With the engine lathe located near the loco-

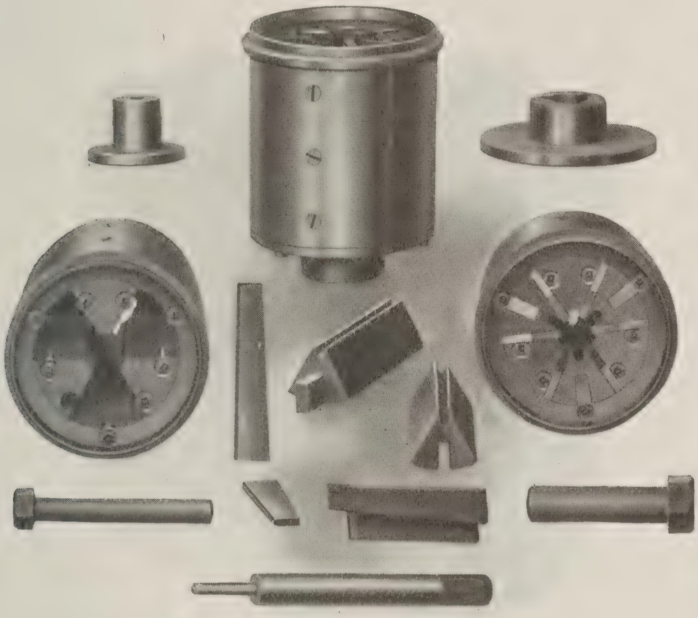

FIG. 5 CUTTER HEAD AND ATTACHMENTS

motive being repaired, the bolts were fitted to the hole after the least possible amount of reaming had been done that would clean up the hole.

19 The improved system contemplates the turning, facing under the head, and placing in stock of standard sizes in lengths of 6, 9, 12, 15, and 18 in. and varying in diameter under the head by thirty-seconds of an inch. Stock may be kept in sixty-fourths of an inch if desired, but very few holes will be found which require less than thirty-

seconds of an inch to clean up. In fact, the chief reason for carrying the intermediate sizes would be to save the hole when it cannot be cleaned up within the next thirty-second. Standard reamers are used, with collars or marks to indicate when they have been driven to the required depth. All bolts have standard hexagon heads conforming to the thread diameter.

20 Bolts are specified with relation to the length and the diameter under the head, and the stock size next longest is used. Under these conditions not more than 3 in. must be cut off to bring the bolt to the proper length. The stock bolts are then taken to the bolt-altering machine, which is a quick-acting hand machine equipped with collet chucks and split bushings for the various diameters of the bolts. The end may be cut off to the proper length and turned for cotter pins, and the head changed to counter sink, box head, button head, or whatever may be required. After threading on the bolt cutter, the bolt is ready to drive in place without further fitting.

21 A comparison of costs by the two methods, taking a $1\frac{1}{8}$ in. × 9 in. bolt as an average would be about as follows:

ENGINE LATHE PRACTICE

	Cost per hundred
Drilling for centers	$0.22
Turning in lathe	2.50
Altering in lathe	$2.50 to 3.50
Threading in bolt cutter	0.22
Cutting off center points	0.10

PRESENT PRACTICE

Pointing the blank	$0.12
Turning by the method described	0.45
Cutting off and changing points and heads where necessary on the bolt-altering machine	$0.40 to 0.60
Threading in the bolt cutter	0.22

22 A device is now being perfected by which the threading can be done automatically at the same time the turning is done. This not only eliminates the bolt cutter charge of $0.22 per hundred, but assured a full, uniform thread absolutely in line with the body of the bolt and square with the facing under the head. When used in connection with a nut faced square with its thread the most satisfactory bolt is obtained.

23 A combined turning and threading device implies a modified form of the cutter head previously described, underneath which is

TABLE OF STOCK SIZES

Showing Eight Threaded Diameters of Bolts and Thirty-Two Diameters Under the Head

Thread Diameter	$\frac{3}{4}$				$\frac{7}{8}$				1				$1\frac{1}{8}$			
Diameters under head	$\frac{25}{32}$	$\frac{13}{16}$	$\frac{27}{32}$	$\frac{7}{8}$	$\frac{29}{32}$	$\frac{15}{16}$	$\frac{31}{32}$	1	$1\frac{1}{32}$	$1\frac{1}{16}$	$1\frac{3}{32}$	$1\frac{1}{8}$	$1\frac{5}{32}$	$1\frac{3}{16}$	$1\frac{7}{32}$	$1\frac{1}{4}$
Length under head......	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
	....	....	....	....	9	9	9	9	9	9	9	9	9	9	9	9
	....	....	....	....	....	....	....	....	....	12	12	12	12	12	12	12
	....	....	....	....	....	....	....	....	....	....	....	....	15	15	15	15
	....	....	....	....	....	....	....	....	....	....	....	....	....	....	18	18
Thread diameter	$1\frac{1}{4}$				$1\frac{3}{8}$				$1\frac{1}{2}$				$1\frac{5}{8}$			
Diameter under head	$1\frac{9}{32}$	$1\frac{5}{16}$	$1\frac{11}{32}$	$1\frac{3}{8}$	$1\frac{13}{32}$	$1\frac{7}{16}$	$1\frac{15}{32}$	$1\frac{1}{2}$	$1\frac{17}{32}$	$1\frac{9}{16}$	$1\frac{19}{32}$	$1\frac{5}{8}$	$1\frac{21}{32}$	$1\frac{11}{16}$	$1\frac{23}{32}$	$1\frac{3}{4}$
Length under head......	6	6	6	6	6	6	6	6	....	....	....	....	....	....	....	....
	9	9	9	9	9	9	9	9	....	....	....	....	....	....	....	....
	12	12	12	12	12	12	12	12	12	12	12	12	....	....	....	....
	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15
	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18

attached a die head of special construction. This die head is carried on four or more vertical rods or guides which are attached to a ring to which the cutter head is fastened. Provision is made for squaring the die head with the cutter head at the time it begins cutting the thread, and at the same time automatically placing the die head in a position where it is free to move in a vertical plane up or down in exact proportion to the difference between the feed and the pitch of the thread to be cut. An automatic knock-out is provided which opens the die head and passes to one side, allowing the threaded bolt to go through to any length within the feed of the machine. Under these conditions it will be seen that so long as the length of the thread to be cut is the same, the length of bolt to be turned is immaterial. The device is very simple in its construction and does not call for a skilled mechanic to adjust or operate it.

TWO PROPOSED UNITS OF POWER

By Prof. Wm. T. Magruder, Columbus, O.

Member of the Society

James Watt is said to have defined a "horsepower" as 33,000 foot-pounds of work per minute, and a "boiler horsepower" as the evaporation of a cubic foot (62 lb.) of water per hour. His rule is sometimes put into the form that "one square foot of grate surface, one square yard of heating surface, a half of a square yard of water surface, and one cubic yard of contents, equals one horsepower, and will evaporate one cubic foot of water per hour in a waggon boiler."

2 Charles E. Emery, Charles T. Porter and Joseph Belknap, "Committee on Boiler Trials of the Judges of Group XX," reported through Horatio Allen, Chairman of Group XX, to Prof. Francis A. Walker, Chief of Bureau of Awards of the United States Centennial Commission of the International Exhibition of 1876, that "the estimated Horse-Power of the several boilers" was given "on the basis that the evaporation of thirty pounds of water is required per horse-power per hour, the results being derived from evaporation at steam pressure of 70 pounds from temperature of 100°."[1] In the Report of the Committee of Judges of Group 20, p. 131, as published by J. B. Lippincott & Co., Philadelphia, "the commercial horse-power of a boiler is fixed at 30 pounds of water evaporated at 70 pounds gage pressure from a temperature of 100 deg."[2] It is to be noted that the time element is omitted. This is not an unusual mistake in speaking of rates, the time element being understood, or taken for granted. This definition is commonly modified so that the Centennial standard of horsepower or the "Centennial horsepower" is defined as the

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

[1] United States International Exhibition, 1876. Reports and Awards, vol. 6, p. 426, sect. 35.

[2] Trans. Am. Soc. M. E., vol. 21, p. 84. Report of Committee on Revision of Standard Code.

"evaporation of thirty pounds per hour of water at 100 deg. fahrenheit into dry steam at 70 pounds gage pressure, or the equivalent." This last phrase is very generally omitted. It is interesting to note that the committee called it the "commercial horsepower," and not a "boiler horsepower." They also defined the "unit of evaporation" as one pound of water at 212 deg. fahr. evaporated into steam of the same temperature, and as being equivalent to 965.7 heat units.

3 In the Appendix is given a summary of the reports of committees of the Society relating to horsepower units and of various discussions on the subject. From these it will be seen that while trying to keep to the Centennial standard of commercial (boiler) horsepower, the committees have gradually veered to the thermal-unit standard, and from the standard of 30-lb. from 100 deg. fahr., into steam at 70-lb. gage pressure.

4 Since 1876, the accuracy of our knowledge of the heat of steam has increased. This is especially true since 1899, the time of the last report to the Society. The "confusion to practical boiler owners," which Dr. Chas. E. Emery seemed[3] to fear might result from the practice of measuring the power of a steam boiler in heat units, does not seem to have materialized.

5 The publication of the new eighth edition of Professor Peabody's Tables of the Properties of Steam, and the publication of the Tables and Diagrams of the Thermal Properties of Saturated and Superheated Steam by Professors Marks and Davis, have complicated this matter still more, and especially with engineering students.

6 According to the steam tables of Charles T. Porter, and the various reports that have been referred to on this subject, the value of the "unit of evaporation" is 965.7 B.t.u. According to Peabody, its value has been gradually changing to 965.8, 966.3, and now to 969.7. According to Marks and Davis, its value should be 970.4. These differences amount to only 4.7 B.t.u. in 970.4, or to one in 205, which is one-half of one per cent. It would seem desirable to use 970 hereafter, instead of 966, as the unit of evaporation, this being the average of the most accurate determinations of the latent heat of evaporation of water at 212 deg. fahr.

7 Similarly, the "unit of commercial evaporation" has been changing from 1110.2 B.t.u. in 1876 and 1884, to 1115.0 according to Peabody, and to 1115.6 B.t.u. according to Marks and Davis today.

[3] Trans. Am. Soc. M. E., vol. 6, p. 334. Report of Committee on Revision of Standard Code.

8 When measured in thermal units, the value of the boiler horsepower, 34½ units of evaporation, is given as 33,305 B.t.u. by the Centennial judges and by the committee reporting to the Society in 1884; as 33,317 B.t.u. in the report of the committee as made in 1899; as 33,320 B.t.u. in one text book on steam-boilers; as 33,454.7 B.t.u. (34½ x 969.7) by Peabody; and as 33,478.8 B.t.u. (34½ x 970.4) by Marks and Davis.

TABLE 1 DIFFERENT VALUES OF A BOILER HORSEPOWER IN B. T. U.

	Units of Evaporation		Units of Commercial Evaporation		B. T. U.
	One	34½	One	30	
Centennial	965.7	33,317	1110.2	33,306	33,305
Peabody	969.7	33,455	1115.0	33,450	....
Marks and Davis	970.4	33,479	1115.6	33,468	....

9 It must be evident to everyone that a would-be standard which has so many different thermal values and is capable of acquiring others with each change in the steam tables is not only indefinite but confusing. It is not a definite unit of measurement, which all standards should be. It seems a pity that in the definition of such a commonly used engineering term there should be any possible chance for confusion and misunderstanding on the part of the student, or for litigation between contractors over the accuracy of the fulfillment of the terms of the contract.

10 Again, for over thirty years, engineers and engineering teachers have been apologizing for the use of the term "boiler horsepower." Even the committee of the Society which reported in 1884, says,[4] "It cannot properly be said that we have any natural unit of power for rating steam boilers." If a horsepower is the rate of doing work, and a boiler is considered as a machine, and the water as the moving parts, the only mechanical power that a boiler produces is that due to the external latent heat of evaporation, except when it explodes. Hence the term "boiler horsepower" is a misnomer. The object of the use of a boiler is the absorption of the heat energy obtained from the potential energy of the fuel by combustion, and the transfer to and

[4] Trans. Am. Soc. M. E., vol. 6, p. 263. Report of Committee on Revision of Standard Code.

storage of the same by a volatile liquid for convenient use in a heat engine, or for other thermal purposes. Hence as a boiler uses the latent heat energy of the fuel as its source of supply, and develops and delivers available heat energy, there would seem to be every reason why the power or ability of a boiler to deliver energy should be measured in thermal units, as being the only unit of energy that the boiler ever normally receives or delivers. Furthermore, the energy from every boiler is always measured in heat units before being reduced to boiler horsepower.

11 To measure the capacity or power of a boiler plant, or its output of energy, in millions of thermal units would not be practical; a smaller unit is desirable. It is therefore proposed to measure the power or capacity of a boiler in "boiler-powers," and to define a boiler-power as 33,000 B.t.u. of heat energy delivered per hour by a steam-boiler, steam main, or by a hot-water heating main, or the like, or added per hour to the feed-water of a boiler, or to the water of a hot water heating system. The acceptance of this term will, it is thought, simplify the whole subject; the unit will remain constant, will be easily remembered and easily used, and will not be one of three standards, differing slightly among themselves, as is at present the case with the term boiler horsepower. Its analogy to mechanical horsepower will be helpful rather than the opposite, especially to the beginner in engineering knowledge. The unit boiler horsepower may still be retained by those who may prefer to use it in some one of its many thermal values.

12 The rapid introduction of gas engines using blast furnace, coke oven, or producer gas, leads to the suggestion of a new unit for the capacity or power of a gas producer, coke oven, or blast furnace, to deliver available heat energy for use in gas engines, under stoves and boilers, or for other thermal uses.

13 At the St. Louis meeting of the American Association for the Advancement of Science in December 1903, the writer read a paper suggesting the term "producer horsepower" as a unit. Since then the question has arisen as to why the old misnomer of "horsepower" should be perpetuated as a unit of measurement of heat energy. Why not simplify and shorten the term "producer horsepower" to "producer power?" If such a unit is desirable for the measurement of the capacity or power of a gas producer, why not suggest similar ones for other generators of heat energy available for use in gas engines and for other thermal uses? Instead of measuring the power of a gas producer in producer powers, and the powers of a blast furnace and

of a coke oven to generate heat energy in blast-furnace powers and coke-oven powers, it is proposed to include all such sources of power, and to measure the heat energies of gaseous and liquid fuels, in "gas powers," and to define a gas-power to be 10,000 B.t.u. of heat energy delivered per hour by a gaseous or liquid fuel. The calorific value should be measured from and to 62 deg. fahr., and at 30 in. of mercury. This unit can be applied and used in the measurement of the energy delivered by a gas well, a gas main, a gas producer, a blast furnace, a coke oven, an oil well, or a pipe line.

14 The number, 10,000 B.t.u., has been chosen as the average in the best gas-engine practice today of heat energy required to develop a horsepower of mechanical energy. The figure bears to current gas-engine practice about the same relation that 30 lb. per hr. of steam at 70 lb. gage pressure from water at 100 deg. fahr. did to current steam-engine practice in 1876. The definition as given contemplates using only the higher calorific value of the fuel, rather than the lower or, so-called, effective value.

15 It is to be hoped that some such unit for the measurement of the output of a generator of heat energy in gaseous or liquid form can be found, and adopted by common consent, before practice and commercial custom in different portions of the country shall have learned to use units which have been less carefully selected and less accurately defined.

APPENDIX

In a paper presented before the Society by Wm. Kent,[1] at the Pittsburg Meeting in May 1884, he tabulates the "horsepower developed at 30 lb. of water evaporated per hour from and at 212°. (Page 268)." In a footnote we read, "The customary method of rating horsepower is 30 lb. of water per horsepower per hour from a feedwater temperature of 212° into steam at 70 lbs. pressure above the atmosphere, which is equal to 30.985 lbs. from feed at 212° into steam of the same temperature. The writer prefers the calculations both of economy and horsepower to be made on the basis of evaporation from and at 212°, for the sake both of uniformity and of convenience in calculation."

2 In a paper presented before the Society by Dr. Chas. E. Emery at the same meeting,[2] he defines the Centennial horsepower (C.H.P.), as being "thirty Kals per hour;" and a "Kal" as being "one pound of water evaporated into saturated steam at seventy pounds pressure from temp. of 100°, with a thermal value of 1110.2 thermal units (Page 282)." This would make a Centennial horsepower equivalent to 33,306 B.t.u. In discussing this paper, Mr. Kent argued that "in determining the horsepower in a steam boiler . . . we should start with the British thermal unit as a basis . . . The unit of evaporation should be 965.7 thermal units. It is the evaporation of one pound of water from and at 212 degrees. A horsepower should be a definite number of units of evaporation—say 30 (Page 297)." He was followed by Dr. E. D. Leavitt, Jr., who thought that "the simplest proposition was to come down to thermal units." Dr. Emery assented that any unit proposed must be based on the "heat unit." In the Centennial Report, this amount of heat energy (1110.2 B.t.u.) was termed the "unit of commercial evaporation (Page 300)."

3 In the Report of the Committee on a Standard Method of Steam-Boiler Trials made at the New York Meeting of the Society in November 1884,[3] the Centennial unit of boiler-power is stated as "30 pounds of water evaporated into dry steam per hour from feed-water at 100° Fahrenheit, and under a pressure of seventy pounds per square inch above the atmosphere." "The quantity of heat demanded to evaporate a pound of water under these conditions is 1110.2 British thermal units, or 1.1496 units of evaporation. The unit of power proposed is thus equivalent to 33,305 heat-units per hour, or 34.488 units of evaporation." Another standard unit for the power of a boiler which was suggested to the Committee in 1884 was "the evaporation of thirty pounds of feed-water into dry steam from and at the boiling point at mean atmospheric pressure (212°F.) (Page 265)." This would have then been taken as the equivalent of 30 x 965.7 = 28,971 B.t.u. per hr. It was not accepted.

[1] Trans. Am. Soc. M. E., vol. 5, p. 260. Rules for Conducting Boiler Tests.
[2] Trans. Am. Soc. M. E., vol. 5, p. 282. Estimates for Steam Users.

4 This Committee recommended in 1884 the adoption of the Centennial Standard and that, for standard trials of steam boilers, the "commercial horsepower be taken as an evaporation of 30 pounds of water per hour from a feed-water temperature of 100° Fahr. into steam at 70 pounds gauge pressure, which shall be considered to be equal to 34½ units of evaporation; that is, to 34½ pounds of water evaporated from a feed-water temperature of 212° Fahr. into steam at the same temperature. This standard is equal to 33,305 thermal units per hour (Page 266)." A footnote gives the "evaporation of 34½ pounds from and at 212°F., as being equal to 30.010 pounds from 100°F., into steam at 70 pounds pressure," and "the 'unit of evaporation' as being 965.7 thermal units," according to the tables in Porter's Treatise on the Richards Steam Engine Indicator, which was the standard of that day.

5 Dr. Chas. E. Emery stated[3] that the "commercial horsepower of 34½ units of evaporation per hour is, for all practical purposes, equal to 33,333 thermal units per hour making it convenient to obtain the horsepower by multiplying the total number of thermal units derived from the fuel per hour by 0.00003 (Page 304)."

6 Prof. J. B. Webb in speaking about the definition of a "commercial horsepower" said "To my mind it would be simpler and better to express results in *thermal units per hour*, and at all events not to express them in horsepowers which are very far from being horsepowers (Page 322)." Nothing came of his suggestion.

7 During this discussion, Wm. Kent introduced and used the term "boiler horsepower" rather than "commercial horsepower," and quoting the definitions of commercial horsepower as recommended by the committee, added, "This standard is certainly not open to the charge of want of exactness and precision (Page 324)."

8 The Committee in its report said that it had "concluded to recommend thirty pounds as the unit of boiler-power (Page 264)".

9 Dr. Charles E. Emery stated that "it was informally suggested to make the standard exactly 33,000 British Thermal Units per hour, so that it would be numerically the same as the number of foot-pounds per minute constituting an actual horsepower, and again 33,333 B.t.u. were suggested to facilitate the calculations, but the general feeling of the committee was against any change whatever (Page 333)." He adds, and seems to prefer the statement that "The value of the unit of horsepower announced is 33,305 British Thermal Units per hour, which being stated in the Report definitely fixes the standard. It also equals 34½ units of evaporation, within one-thirtieth of one per cent."

10 Prof. W. P. Trowbridge, in discussing the report, called attention to the diversity of opinion in the committee as to whether the "unit of boiler-power" should be expressed in terms of the "unit of evaporation," or in some other terms.

11 Prof. W. P. Trowbridge and Prof. C. B. Richards presented a paper at the Boston meeting in 1885 on The Rating of Steam Boilers by Horse-Powers for Commercial Purposes,[4] in which they differ from the committee which had reported the preceding year, but quote its report with the statement "What is

[3] Trans. Am. Soc. M. E., vol. 6, p. 256.
[4] Trans. Am. Soc. M. E., vol. 7, p. 214.

needed is a standard unit of boiler power which may be used commercially in rating boilers, and in specifications presenting the power to be demanded by the purchaser and guaranteed by the vender (Page 216)."

12 George H. Babcock in discussing this subject said, "A dynamic horsepower in its simplest form is 33,000 foot-pounds per minute. A boiler horsepower should be defined as 33,000 heat units per hour imparted to the water (Page 225)." Mr. Kent stated, "The term horsepower has two meanings in engineering literature: First, an absolute unit or measure of the rate of work, ... and an approximate measure of the size, capacity, value, or "rating," of a boiler engine, water-wheel or other source or conveyor of energy, by which measure it may be described, bought and sold, etc. (Page 226)."

13 In the Report of the Committee on the Revision of the Standard Code for Conducting Steam-Boiler Trials, made in 1899,[5] it is stated (Page 36) that "The Committee approves the conclusions of the 1885 Code to the effect that the standard 'unit of evaporation' should be one pound of water at 212 degrees Fahr. evaporated into dry steam of the same temperature. This unit is equivalent to 965.7 British thermal units. The Committee recommends that, as far as possible, the capacity of a boiler be expressed in terms of the 'number of pounds of water evaporated per hour from and at 212 degrees.' It does not seem expedient, however, to abandon the widely recognized measure of capacity of stationary or land boilers expressed in terms of 'boiler horsepower.' The present committee accepts the same standard, but reverses the order of the two clauses in the statement, and slightly modifies them to read as follows: 'The unit of commercial horsepower developed by a boiler shall be taken as $34\frac{1}{2}$ units of evaporation per hour; that is, $34\frac{1}{2}$ pounds of water evaporated per hour from a feed-water temperature of 212 degrees Fahr. into dry steam of the same temperature. This standard is equivalent to 33,317 British thermal units per hour. It is also practically equivalent to an evaporation of 30 pounds of water from a feed-water temperature of 100 degrees Fahr. into steam at 70 pounds gauge pressure.'" In a footnote is added the statement that "The unit of evaporation being equivalent to 965.7 thermal units, the commercial horsepower = 34.5 x 965.7 = 33,317 thermal units (Page 37)."

[5] Trans. Am. Soc. M. E., vol. 21, p. 34.

GAS ENGINES FOR DRIVING ALTERNATING-CURRENT GENERATORS

H. G. Reist, Schenectady, N. Y.

Member of the Society

The problem of driving an alternating-current generator by means of a gas engine is not inherently different from that of driving it from a steam engine. If the shaft of the engine turned with a uniform motion, no difficulty would be experienced and no special design would be necessary. It is the variations in angular velocity and speed that affect the driving of alternators.

2 If the current of a single generator is used for lights or for heating, as in electric furnaces or in electrolytic work, variations in velocity either during a single turn or due to the hunting of the governor will simply increase and decrease the load as the speed varies. If induction motors are driven from a single generator, it is only under peculiar circumstances that any trouble is experienced due to speed variations in the engine, because this type of motor is asynchronous and does not have to follow exactly; it is as if it were belted to the engine, the connection being slightly flexible.

3 A synchronous motor or rotary converter, on the other hand, must keep in phase and behave as though geared to the engine, and must respond to all its speed variations. If it does not keep absolutely even, that is, if its phase relations change, cross currents will flow. When two or more generators are operated in parallel, their behavior is similar, any angular departure of one from the other causing a cross-current. The volume of the cross-current depends, with any given design of generator, on the angular departure of the generators from each other. This departure may be twice the angular variation of the engine rotating parts from a mean position, because one may be a maximum distance ahead while the other is in the most backward position.

The American Society of Mechanical Engineers, 29 West 39th Street, New York, June 1910. All papers are subject to revision.

4 If the generators were mounted on the shaft so that the relations of the poles to the cranks were identical, and if it were possible so to synchronize them that the corresponding cranks of engines to be run together were exactly together, no cross-currents would flow, because the engines would slow down and speed up together. This is not feasible, however, and it becomes necessary to design the engine to run with a fairly uniform rotation. It has been found good practice to limit the variation from a mean position of the revolving parts of the electric generator to $1\frac{1}{4}$ electrical degrees.

5 An electrical degree is 1-360 part of the space occupied by two poles on a generator; that is, a two-pole generator is the unit and an electrical degree is one mechanical degree of such a machine; if the generator has four-poles, an electrical degree will be one-half a mechanical degree of the circle on which the poles are mounted; if there are 6 poles, it will be one-third of a mechanical degree. In general, to reduce electrical degrees to mechanical degrees, we must divide the allowable variation by one-half the total number of poles on the generator; so that the $1\frac{1}{4}$ electrical degrees mentioned above for a twenty-pole machine would be 0.125 actual degrees on the circumference of the flywheel. From this it is evident that with a generator of many poles, a more even speed is needed than for one with few poles. For a 60-cycle generator, which at a given engine speed has $\frac{60}{25}$ as many poles as a 25-cycle generator, the evenness of running must be much greater than for a 25-cycle generator.

6 The cross-currents between two electrical generators tend to speed up the lagging machine, bringing them more closely into synchronism. If there were no inertia the rotating parts of generator and flywheel would quickly get into synchronism, reducing and almost eliminating the cross-currents. This is, however, an ideal condition. The cross-currents are a factor of the amount of inertia with a given natural angular variation, and it will readily be seen that from this standpoint the larger the flywheel the less effect a given value of currents or torque will have on the mass. If the flywheel is very large, the currents which it may be practical to allow to flow between the machines may not be able to draw them together at all. Hence a large flywheel, while useful in obtaining uniform rotation, so far as the engine is concerned, prevents the current flowing between the machines from being very effective in drawing them into synchronism. This shows that it is desirable to obtain uniform rotation in other ways than by the use of an excessively heavy flywheel. Currents flowing between machines occasion losses in the copper and this

adds to the heating of the machine. They thus reduce the output with a given rise of temperature and reduce the efficiency.

7 Certain elements of design may be introduced into an electrical generator to make it less sensitive to slight variations in turning moment supplied by the engine, such as building a generator of poor regulation. The regulation must not be too poor, however; otherwise the operation of the system will be unsatisfactory. A "squirrel-cage" winding in the poles of the generator allows secondary currents to flow in this part of the structure and increase the torque, tending to draw the generators together with a given interchange of current between the two machines. This is of great assistance in parallel operation of generators and should generally be applied on generators to be driven by gas engines. If a flexible connection could be provided between engine and generator it would greatly assist in satisfactory parallel operation, but this connection is generally applicable only on small machines. The ultimate solution lies in the direction of greater uniformity of motion in the engine itself.

8 Uniformity of rotation of gas engines is dependent on a number of elements of design, such as (*a*) the number of impulses per revolution, which in turn is dependent on the number of cylinders and arrangement of cranks and on whether a two or a four-cycle system is used; (*b*) the compression and weight of the reciprocating parts; (*c*) the time of ignition; (*d*) the weight of the flywheel. The use of a heavy flywheel, however, while one of the simplest, is the least desirable method of obtaining even rotation of the engine shaft, and other means should be used to obtain as uniform rotation as possible.

9 The following seem to be the desirable characteristics of gas engines for driving alternators:

a High speed. This will require fewer poles with a given frequency and a greater angular variation will be allowable.

b A light flywheel. This will allow the current to keep the generators together with a minimum disturbance.

c Large engines should be built with many cylinders and cranks so placed as to contribute to an even turning moment.

CRITICAL SPEED CALCULATION

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Non-Member

Critical speed is the term applied to the speed of a rotating body at which occur the maximum vibrations of the revolving mass or supporting structure. The vibrations are smaller for speeds both above and below the critical value. Hence the importance, to the designer of high rotative speed apparatus, of predetermining these maximum vibrating points. The high speeds and large capacities now being used in electrical machinery, such as turbo-generators, frequency changers, etc., bring this apparatus within the critical-speed range; and the electrical designer must study the vibrating properties of his high speed machines, or leave this operating trouble to chance.

2 The phenomenon of critical speed was known to De Laval, who designed his turbines with a small or "flexible" shaft, so that the running speed was seven to ten times the critical value. So far as is known he did not understand the mathematical theory.

3 The first scientific explanation of critical speed is due to Rankine who in Machinery and Millwork gave the mathematical explanation for a shaft with its own weight only, when supported at each end, and also for fixed direction at one end as a cantilever. This was followed by Professor Greenhill[2] with an explanation for an unloaded shaft with fixed direction at each end. Professor Reynolds[3] then extended the mathematical treatment to shafts loaded with pulleys, and Professor Dunkerley[3] proved the formulæ by laboratory experiments and developed an approximate formula for shafts with more than one load. Reynolds and Dunkerley do not satisfactorily treat

[1] General Electric Company, Schenectady, N. Y.

[2] Proceedings Institution of Mechanical Engineers, April 1883.

[3] Philosophical Transactions, Royal Society, London, vol. 185a, 1895; Proceedings, Liverpool Engineering Society, 1895.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

the case of two loads on the shaft and their method was criticised by Dr. Chree.

4 Föppl[1] in Germany gave the case for a single concentrated load of the shaft. Stodola[2] in 1903 first gave the formula for any two concentrated loads on a shaft. Professor Morley[3] has lately given approximate formulæ for combined distributed and concentrated loads.

5 These constitute practically all of the literature on the subject. They are mainly mathematical demonstrations and do not leave the subject in convenient form for the use of the designing engineer. This paper will give a mathematical treatment for both the distributed and the concentrated loads, by considering the motion of the shaft as vibratory along two axes, study the vibrations for all speeds, reduce the formulæ to practical form, and present them in tables for convenient use.

NATURE OF CRITICAL SPEED

6 To explain more easily the nature of critical speed we will first give the simple solution of Föppl for a single load. All critical-speed calculations assume an unbalanced load. It is practically impossible to balance a rotating mass so that its center of gravity exactly coincides with the mechanical axis of rotation. As the mass starts to rotate, the center of gravity will rotate in a very small radius around the shaft center. The rotation of the center of gravity at this small radius produces a centrifugal force which acts radially outward from the shaft center through the center of gravity, and rotates around the shaft with the center of gravity. Consider the case shown in Fig. 1, of a single concentrated load on a vertical shaft. Let a be the unknown distance from the center of gravity of the mass to the center of the shaft. The centrifugal force of this mass m, rotating at the radius a, will tend to deflect the shaft in the direction of a, so that the shaft will rotate in a bowed condition. The bowed shape will in itself increase the circle in which the center of gravity rotates; this increases the centrifugal force, and in turn the shaft deflection. This action continues until finally a state of equilibrium is reached where the force of the shaft deflection is equal and opposite to the centrifugal force of the mass. This condition of equilibrium is shown

[1] Civil-Ingenieur, 1895, p. 333.
[2] The Steam Turbine, Stodola, p. 183.
[3] Engineering (London), 1909, vol. 88, p. 135.

in Fig. 2, where the center of gravity is rotating at the radius r, and the shaft rotating in a bowed condition at the radius or deflection $(r\text{-}a)$. Let $\Delta = \frac{W\ K}{E\ I}$ – static deflection of shaft, if horizontal, and p (angular velocity) $= \frac{2\ \pi}{60\ N}$, where N = r. p. m. The centrifugal force of the center of gravity is mrp^2. This centrifugal force would produce a deflection of $m\ r\ p^2 \frac{K}{E\ I} = r\ \ p^2 \ \frac{W\ K}{g\,E\,I} = r\ p^2 \frac{\Delta}{g}$, where g is gravity. But the shaft deflection opposing the centrifugal force is, for equilibrium $(r\text{-}a)$. This gives the equation

$$r - a = r \frac{p^2\ \Delta}{g}$$

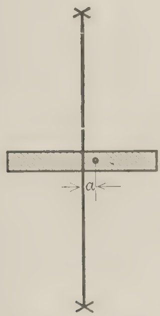


FIG. 1 CONCENTRATED LOAD ON VERTICAL SHAFT AT REST

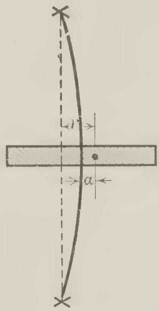


FIG. 2 CONCENTRATED LOAD ON VERTICAL SHAFT IN MOTION

which solved for r gives

$$r = \frac{\frac{ga}{\Delta}}{\frac{g}{\Delta} - p^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

7 This equation, with values of r plotted against p, is shown in dotted lines in Fig. 3. As the angular speed p increases from zero, the radius or deflection r increases, until r becomes theoretically infinite when

$$p^2 = \frac{g}{\Delta}$$

This is the condition of maximum vibration produced by the shaft; and the critical number of revolutions is found from the equation

$$p^2 = \frac{g}{\Delta}$$

$$\left(\frac{2\pi}{60}N\right)^2 = \frac{32.2 \times 12}{\Delta}$$

$$N = \frac{187.7}{\sqrt{\Delta}} \qquad (2)$$

for inch, pound, minute, units.

8 Referring to the curve beyond the critical-speed value, r becomes negative, and as the value of p is increased r approaches the

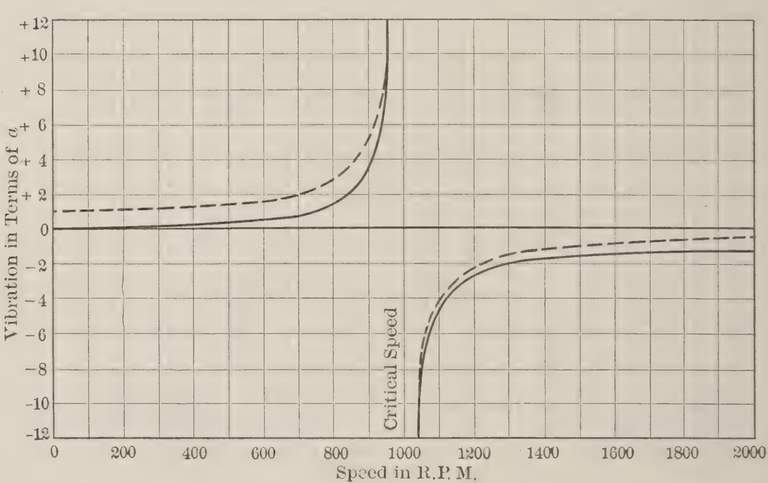


Fig. 3 Amplitude of Vibration With Single Concentrated Load (Figs. 2 and 4).

DOTTED LINE FOR EQUATION 1. SOLID LINE FOR EQUATION 4

limit of zero; in other words, above the critical speed the center of gravity revolves inside the bow of the shaft, or in a smaller circle than the shaft center; and the tendency of the rotating mass is to rotate about its own center of gravity, and not about the mechanical center. It approaches its center of gravity as a limit for infinite speed.

9 The natural time of vibration of a loaded shaft is

$$t = 2\pi\sqrt{\frac{\Delta}{g}}$$

and the number of natural vibrations per minute is

$$\frac{60}{t} = \frac{60}{2\pi}\sqrt{\frac{g}{\Delta}}$$

which is the same as N in Equation 2, the critical number of revolutions. Thus for a single concentrated load the critical-speed phenomena occur when the revolutions synchronize with the natural period of vibration of the shaft. No satisfactory explanation has been given of the detail action at the critical speed, or of the manner in which the center of gravity passes from the outside to the inside of the bow of the shaft. Theoretically the deflection or bow of the shaft becomes infinite at the critical speed. Practically it does not, because of the resistance of the air and probably the need of the factor of time to accumulate energy.

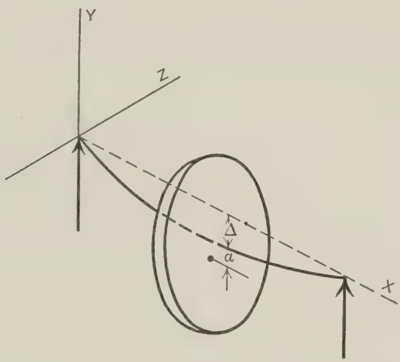


Fig. 4 Concentrated load on Horizontal Shaft

10 In machines where the normal running speed is higher than the critical speed, the shaft is made just strong enough to withstand the deflection in passing through the critical speed, and as weak or flexible as possible for the smooth running above the critical speed. The weaker the shaft, the lower the critical speed, the nearer approach to rotation about the center of gravity, and the less bow or deflection in the shaft.

11 This solution is satisfactory so far as the critical value and deflection of the rotating mass is concerned; and it affords a simple explanation of the actions of a rotating body. But in the design of a machine the vibrations of the frame or supporting structure are of equal or greater importance. The shaft rotating in its bowed condition has a reaction on the bearing points, the reaction rotating with

the shaft. This force is the impressed vibration that causes the frame to vibrate. If we determine the shaft deflection during rotation, or the location of the shaft axis at any instant, we can find the amount of the force of the shaft, or the impressed vibration on the frame.

12 When coördinate axes, as shown in perspective in Fig. 4, are taken, and the location and motion of the shaft center at any instant are determined, the force impressed upon the frame is measured by the coördinates of the shaft center. If we sum the forces along each axis, the solution gives us a form of compound harmonic vibration. This same method affords a comparatively easy algebraic solution for two loads; and is applied equally well to horizontal and vertical shafts

SINGLE CONCENTRATED LOADS

13 For the condition shown in perspective in Fig. 4, Δ is the static deflection at the disc load when the shaft is horizontal, and a the distance from the shaft center to the center of gravity. To simplify the

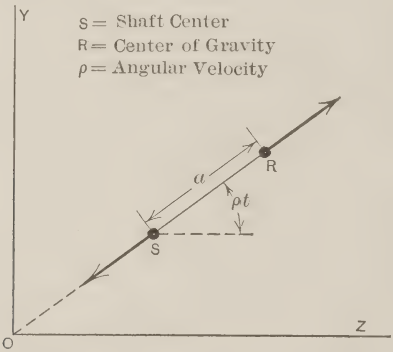


FIG. 5 CONCENTRATED LOAD ON HORIZONTAL SHAFT (FIG. 4)

calculations consider the shaft in a vertical position. It will be later shown that the same formulæ apply to horizontal shafts. Pass the YZ plane (Fig. 5) through the disc perpendicular to the shaft. When at rest the center of the vertical shaft is at the origin O. When in motion, at a given time the shaft center is at S, with the coördinates (yz); and the center of gravity is at R, a constant distance a from S. Due to the turning of the shaft, the point R revolves around S with the angular velocity p, so that the angle turned through is pt.

14 The force acting on the point S is the spring of the shaft towards the zero position. This is $\frac{W}{\Delta}\overline{OS}$ where $\frac{W}{\Delta}$ is the force of the

shaft per unit of deflection. Since the force from S acts towards the origin, for equilibrium of moments about O the centrifugal force of the disc at R must act in a line through the origin. Also for equilibrium of the two forces they must act in line with each other, be equal in value and be opposed in direction. Then for R to turn about S with the angular velocity p, R must revolve about the origin with the same angular velocity, and in a circle with center at O.

15 The centrifugal force acting at R is $mp^2\,\overline{OR}$. The component parallel to the Y-axis is $mp^2\,(y + a \sin pt)$; and to the Z axis is $m\,p^2\,(z+a \cos pt)$. The spring of the shaft acting at S is $\frac{W}{\Delta}\,\overline{OS}$, with a component parallel to the Y-axis of $-\frac{W}{\Delta}\,y$; and to the Z-axis, $-\frac{W}{\Delta}\,z$. The sum of these forces along the Y-axis is

$$mp^2\,(y + a \sin pt) - \frac{W}{\Delta}\,y = O$$

and for the Z-axis

$$mp^2\,(z + a \cos pt) - \frac{W}{\Delta}\,z = O$$

Dividing by m and solving for y and z these equations give

$$y = \frac{ap^2}{\frac{g}{\Delta} - p^2} \sin pt \qquad (3)$$

$$z = \frac{ap^2}{\frac{g}{\Delta} - p^2} \cos pt \qquad (4)$$

INTERPRETATION OF EQUATIONS

16 Equations 3 and 4 determine the motion or path of the shaft center. Taken together they are the equations of a circle with center at the origin. Taken separately, the equations are of the form of simple harmonic motion, with a forced vibration of $a \sin pt$ along the Y-axis, and $a \cos pt$ along the Z-axis. $p = \frac{2\pi}{60}$ times the frequency of vibration. The coefficients of the sine and cosine are the amplitude of the vibration along each axis. These are plotted in full lines in

Fig. 3. The amplitude of vibration, being the same for both axes, contains only the independent variable p. The amplitude will increase as the speed or p increases, until the vibration becomes infinite when $\frac{g}{\Delta} - p^2 =$ zero. This is the same critical-speed condition as in the previous solution, Equation 2. Beyond the infinite value the coefficients become negative and decrease, becoming smaller the higher the speed.

17 This can have the physical interpretation that before the critical speed is reached the center of gravity revolves outside of, or in a larger circle than, the mechanical center of the shaft. Beyond the critical-speed point, the center of gravity rotates inside of, or in a smaller circle than, the shaft center.

18 As previously shown, the critical speed occurs when the rotation synchronizes with the natural period of vibration of the loaded shaft. It may be seen from the curve that when the frequency of the forced vibration $\frac{60}{2\pi} p$ is nearly equal to the frequency of the natural vibration $\frac{60}{2\pi}\sqrt{\frac{g}{\Delta}}$ we have a similar state of things to that which gives rise to *resonance* in acoustic instruments and electrical circuits.

19 The natural period of vibration and the forced vibration are the same for either a vertical or a horizontal position of the shaft, so that the same critical-speed formulæ apply for either position. When vertical, the center of the vibration or of the rotation is at $y = 0$, $z = 0$; when horizontal, the center of the vibration along each axis is at $y = -\Delta$, $z = 0$. The horizontal position is equivalent to a change of coördinate axes from $y = 0$ to $y = -\Delta$, so that Equation 3 becomes

$$y = \frac{ap^2}{\frac{g}{\Delta} - p^2} \sin pt - \Delta \qquad (3a)$$

20 Vibration is caused by an unbalance of the body, or by the center of gravity not coinciding with the mechanical center of the shaft. The centrifugal force of the unbalance causes an accelerating force along each axis, or a forced vibration of a amplitude. This forced vibration causes the shaft to vibrate along each axis with the amplitude of $\frac{ap^2}{\frac{g}{\Delta} - p^2}$. This shaft vibration is in turn the vibration

that is forced on the frame or supporting structure and causes it to vibrate. This latter value is therefore the vibration to be considered in the design of machines. Comparing the two curves of Fig. 3 it will be noted that at zero speed the vibration in Equation 1 is a; in Equation 4 it is zero. Beyond the critical speed, Equation 1 approaches zero; Equation 4 approaches a. This difference is due to Equation 1 considering the motion of the center of gravity, and Equation 4 the motion of the shaft center. Equation 4 is the vibration impressed on the frame and therefore the value to be considered.

TWO CONCENTRATED LOADS

21 Equations covering any two concentrated loads, with either two or three bearing supports, may be developed by the same method as in the previous case. Take the condition shown in Fig. 6, with two discs for concentrated loads and three bearing points. To distinguish

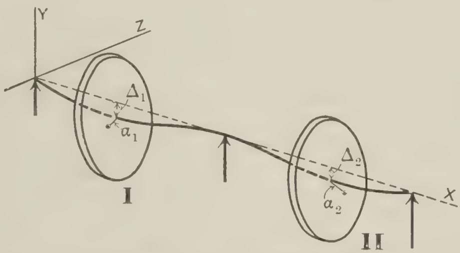


Fig. 6 Two Concentrated Loads on Horizontal Shaft

symbols, let the Y Z plane passing through disc I be represented by sub-letters 1, and the plane through disc II by sub-letters 2. To determine the influence one disc has upon the other the equations $F_1 = K_1 y_1 + K_3 y_2$ and $F_2 = K_2 y_2 + K_3 y_1$ are taken, where y_1 and y_2 are any positions of the shaft deflections on the Y-axis, and F_1 and F_2 are the shaft forces due to the deflections which act toward the unloaded or zero position. K_1, K_2 and K_3 are constants for a given shaft and can be deduced from the deflection equations of beams for different loads and supports. Assume the coördinates of the shaft center in any position y_1, z_1, in the plane through disc I and y_2, z_2 in the plane through disc II. Take the distances from the centers of gravity to the shaft centers to be a_1 and a_2 and their directions to differ by x deg. on the Y Z plane. The forces acting on the discs when the shaft is vertical are:

First, centrifugal force:

Plane I, Y-axis, $m_1 p^2 (y_1 + a_1 \sin pt)$;
Z-axis, $m_1 p^2 (z_1 + a_1 \cos pt)$.
Plane II, Y-axis, $m_2 p^2 [y_2 + a_2 \sin (pt + \alpha)]$;
Z-axis, $m_2 p^2 [z_2 + a_2 \cos (pt + \alpha)]$.

Second, reaction or spring of the shaft:

Plane I, Y-axis, $K_1 y_1 + K_3 y_2$; Z-axis, $K_1 z_1 + K_3 z_2$.
Plane II, Y-axis, $K_2 y_2 + K_3 y_1$; Z-axis, $K_2 z_2 + K_3 z_1$.

The summation of these forces along the axes gives the following equations:

$$m_1 p^2 (y_1 + a_1 \sin pt) - K_1 y_1 - K_3 y_2 = 0.$$
$$m_1 p^2 (z_1 + a_1 \cos pt) - K_1 z_1 - K_3 z_2 = 0.$$
$$m_2 p^2 [y_2 + a_2 \sin (pt + \alpha)] - K_2 y_2 - K_3 y_1 = 0.$$
$$m_2 p^2 [z_2 + a_2 \cos (pt + \alpha)] - K_2 z_2 - K_3 z_1 = 0.$$

The solution of these equations gives:

$$y_1 = A \sin pt - B \cos pt \quad [5]$$
$$z_1 = B \sin pt + A \cos pt \quad [6]$$
$$y_2 = C \sin pt + D \cos pt \quad [7]$$
$$z_2 = -D \sin pt + C \cos pt \quad [8]$$

where

$$A = \frac{(K_2 - m_2 p^2)\, m_1 a_1 p^2 - K_3 m_2 a_2 p^2 \cos \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$B = \frac{K_3 m_2 a_2 p^2 \sin \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$C = \frac{(K_1 - m_1 p^2)\, m_2 a_2 p^2 \cos \alpha - K_3 m_1 a_1 p^2}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

$$D = \frac{(K_1 - m_1 p^2)\, m_2 a_2 p^2 \sin \alpha}{(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2}$$

INTERPRETATION OF EQUATIONS

22 Equations 5 to 8 are of the form of harmonic motion along their respective axes. Equations 5 and 6 taken together (also Equations 7 and 8) are equations of a circle with center at the origin, the radius of the circle being $\sqrt{A^2 + B^2}$.

23 The coefficients which represent the radii in rotation, or the amplitude of vibration, have the same denominators. The value of the coefficients becomes infinite when the denominators equal zero, which is the critical-speed condition. That is,

$$(K_1 - m_1 p^2)(K_2 - m_2 p^2) - K_3^2 = O.$$

$$p = \sqrt{\frac{g}{2}\left[\frac{K_1}{W_1} + \frac{K_2}{W_2} \pm \sqrt{\left(\frac{K_1}{W_1} - \frac{K_2}{W_2}\right)^2 + \frac{2K_3^2}{W_1 W_2}}\right]} = \frac{2\pi}{60} N$$

$$N = 132.3\sqrt{\frac{K_1}{W_1} + \frac{K_2}{W_2} \pm \sqrt{\left(\frac{K_1}{W_1} - \frac{K_2}{W_2}\right)^2 + \frac{2K_3^2}{W_1 W_2}}} \quad \text{.........[9]}$$

for inch, pound, minute units.

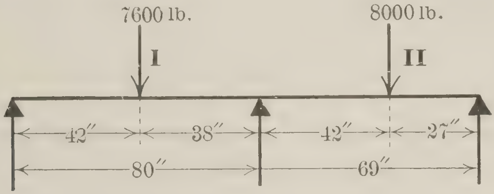


Fig. 7 Condition of Load on Shaft (Fig. 6)

24 The ± sign of this equation gives two values of critical speed. This equation is general for two concentrated loads regardless of the method of support, for either two of three bearings.

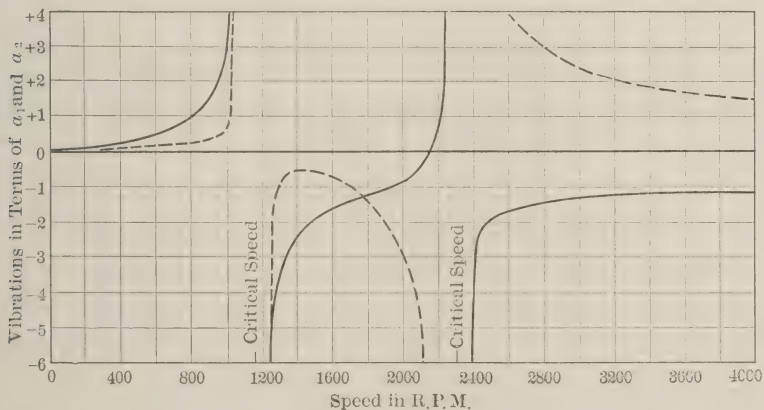


Fig. 8 Amplitude of Vibration with two Concentrated Loads (Figs. 6 and 7)

SOLID LINE FOR LOAD I; DOTTED LINE FOR LOAD II

25 When the unbalances a_1 and a_2 of the centers of gravity of both loads lie in the same plane, either on the same or opposite sides of the shaft, so that the angle α is zero or π, the coefficient B becomes zero and the radius of the circle of the shaft path is A. This gives below the critical-speed value the smallest circle when the unbalances are on the same side, or α equal to zero degrees; above the critical speed, π gives the smallest circle.

26 The properties of Equations 5 and 8 can be shown more fully by an example. For the conditions given in Fig. 7 the amplitude of the vibrations is plotted in Fig. 8. This machine showed excessive vibration between 1100 and 1200 r.p.m., when not in nearly perfect balance. It could not be speeded up to the second critical speed, the second value being too far above the running speed. The solid curve is the vibration of Load I; the dotted curve is the vibration of Load II. The amount of vibration is in terms of the unbalance a_1 and a_2.

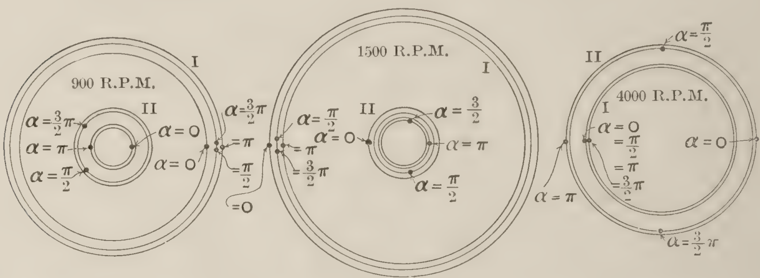


Fig. 9, 10, 11 Path and Relative Location of Shaft Centers for Different Angles of Unbalance

For the same unbalance, Load I, having the weaker portion of the shaft, has the largest vibrations until the first critical speed is reached. Beyond the first critical value the vibrations of Load II become large and influence the vibrations of Load I, reducing them through zero from a negative to a positive value. Beyond the second critical speed, Load II with the stiffer shaft has the larger vibrations.

27 Another interesting thing is the relative location of the shaft centers at any given time. Figs. 9, 10 and 11 show the paths of the shaft centers and their location on the circles, for the angle between the unbalances of $\alpha = 0, \frac{1}{2}\pi, \pi$ and $\frac{3}{2}\pi$, when $pt = 0, 2\pi$, etc. The rotation of the points on all circles is in the same direction as the rotation of the machine. When below the critical speed (Fig. 9),

Load I on the weaker shaft, or the shaft with the largest static deflection, rotates in the larger circle. Note the relative positions of the shaft centers for different values of α; and the influence the larger circles have upon Load II in forcing the unbalances towards opposite sides of the shaft as shown by the positions for $\alpha = \frac{1}{2}\pi$ and $\frac{3}{2}\pi$. Between the two critical speeds (Fig. 10), the positions for all values of α have turned through 180 deg., except Load II, $\alpha = \frac{1}{2}\pi$ and $\frac{3}{2}\pi$. Above the second critical speed the positions of Load II turn through another 180 deg., while Load I is unchanged. Here the stiffer shaft, or Load II, has the larger circles of rotation.

CONSTANTS FOR TWO LOADS

28 As the sixty-nine formulæ given in the table for calculations cannot be derived for want of space, an example will be given to illustrate the method. Take the condition of two loads just considered, using the letters given in Fig. 12 for dimensions and weights. The

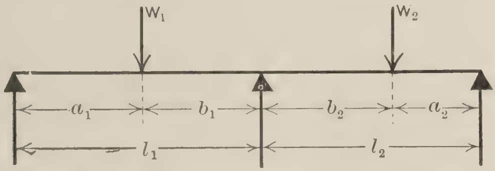


FIG. 12 CONDITIONS OF LOADING

force of the spring of the shaft is $F_1, = K_1 y_1 + K_3 y_2$ at W_1, for any deflections y_1 and y_2; and $F_2 = K_2 y_2 + K_3 y_1$ at W_2 for any position y_1 and y_2. The standard equations for deflections at the loads are:

$$\Delta_1 = \frac{a_1 b_1}{6E\,Il_1}\left[2\,a_1 b_1 W_1 - m\,(l_1 + a_1)\right]$$

$$\Delta_2 = \frac{a_2 b_2}{6E\,Il_2}\left[2\,a_2 b_2 W_2 - m\,(l_2 + a_2)\right]$$

$$m = \frac{1}{2\,(l_1 + l_2)}\left[\frac{W_1 a_1 b_1}{l_1}(l_1 + a_1) + \frac{W_2 a_2 b_2}{l_2}(l_2 + a_2)\right]$$

Making variable by changing Δ_1 to y_1, Δ_2 to y_2, W_1 to F_1, W_2 to F_2, and solving for F_1 and F_2, gives equations of the above form, where

$$K_1 = \frac{C\ l_1^2}{a_1^2\ b_1^2}[4\ l_2\ (l_1 + l_2) - (l_2 + a_2)^2]$$

$$K_2 = \frac{C\ l_2^2}{a_2^2\ b_2^2}[4\ l_1\ (l_1 + l_2) - (l_1 + a_1)^2]$$

$$K_3 = \frac{C\ l_1\ l_2}{a_1\ b_1\ a_2\ b_2}(l_1 + a_1)\ (l_2 + a_2)$$

$$C = \frac{3\ E\ I}{4\ l_1\ l_2\ (l_1 + l_2) - l_1\ (l_2 + a_2)^2 - l_2\ (l_1 + a_1)^2}$$

Constants for other dimensions, loads and supports, may be derived from the deflection formulæ in a similar manner.

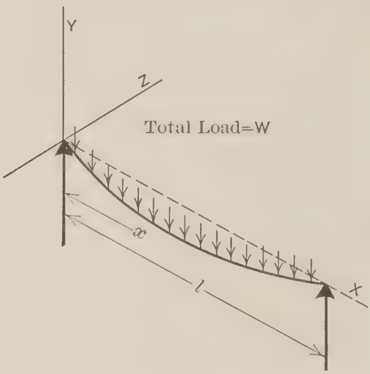


Fig. 13 Shaft with Uniform Load

DISTRIBUTED LOADS

29 All critical speed formulæ so far developed for distributed loads are based on the equation in Mechanics that

$$E\,I\,\frac{d^4\,y}{d\,x^4} = \text{the weight or force } on \text{ a unit length of beam}$$

and there is an unbalance caused by the center of gravity not coinciding with the shaft center.

30 To simplify the calculations, assume the conditions of Fig. 13, of a shaft with constant diameter uniformly loaded over its entire length by a load, as disc wheels, which will not affect the flexibility of the shaft; and that the centers of gravity of all the discs lie in the same plane and on the same side of the shaft, at a constant distance a

from the shaft center, similar to the single disc in Fig. 4. W = total weight of shaft and discs.

31 Taking any unit length along the X axis, its mass is $\frac{W}{l\,g}$. Assume the center of rotation to be at the origin of the axes when the shaft is vertical. The centrifugal force of the mass of unit length is $\frac{W}{l\,g}\,p^2$ times the radius to center of gravity. This radius projected on the Y axis is $(y + a \sin pt)$ and on the Z axis is $(z + a \cos pt)$, where y and z are coördinates of the shaft center for the shaft in any position of rotation.

32 The forces acting on a unit length, projected on the axes, are: First, centrifugal force: Y axis, $\frac{W\,p^2}{l\,g}\,(y + a \sin pt)$; Z axis, $\frac{W\,p^2}{l\,g}$ $(z + a \cos pt)$.

The second force, the spring of the shaft, does not enter as we are considering the forces acting *on* the shaft, and not the reaction of the shaft.

33 The equation in Mechanics of the forces acting on a unit length gives for the Y axis:

$$E\,I\,\frac{d^4y}{dx^4} = \frac{W\;p^2}{l\;\;g}\,(\,y + a \sin pt)$$

and for the Z axis:

$$E\,I\,\frac{d^4\,z}{d\,x^4} = \frac{W\,p^2}{l\,g}\,(z + a \cos pt)$$

The general solutions of these equations are:

$$y = [Fe^{kx} + Ge^{-kx} + H \cos kx + J \sin kx - a] \sin pt \ldots\ldots [10]$$

$$z = [Fe^{kx} + Ge^{-kx} + H \cos kx + J \sin kx\text{-}a] \cos pt \ldots\ldots\ldots\ldots [11]$$

where

$$k = \sqrt[4]{\frac{W\,p^2}{E\,I\,g\,l}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots [12]$$

e = the base of the natural system of logarithms, and the capital letters are constants determined by the conditions imposed on the equations by the supports, etc., as shown in the following special cases.

SHAFT SUPPORTED AT BOTH ENDS

34 A shaft supported at both ends, as shown in Fig. 13, either vertical or horizontal, imposes the conditions of deflection y, and moment $EI\dfrac{d^2 y}{d x^2}$, both equal to zero at the supports, or when x equals zero and when x equals l. This gives four equations with four unknown constants as follows:

For $x = 0, y = 0$

$$0 = F + G + H - a$$

For $x = 0, \dfrac{d^2 y}{d x^2} = 0$

$$0 = F + G - H$$

For $x = l, \ y = 0$

$$0 = Fe^{kl} + Ge^{-kl} + H \cos kl + J \sin kl - a$$

For $x = l, \dfrac{d^2 y}{d x^2} = 0$

$$0 = Fe^{kl} + Ge^{-kl} - H \cos kl - J \sin kl$$

The solution of these four equations gives

$$F = -\frac{a}{2}\left(\frac{e^{-kl} - 1}{e^{kl} - e^{-kl}}\right) \qquad G = \frac{a}{2}\left(\frac{e^{kl} - 1}{e^{kl} - e^{-kl}}\right)$$

$$H = \frac{a}{2} \qquad J = \frac{a}{2}\left(\frac{1 + \cos kl}{\sin kl}\right)$$

These values substituted in the general equations 10 and 11 for distributed loads give values for y and z which represent the path of the shaft center for any point x along the length. The coefficients of $\sin pt$ and $\cos pt$ are the amplitude of vibration along each axis, or the radius of the circle in which any point on the shaft center rotates. These coefficients become infinite or have the critical speed value when $\sin kl = 0$, or whenever $kl = \pi, 2\pi, 3\pi$, etc. Since p is proportional to k^2 by equation 12 we have an infinite number of critical speeds which have the ratio $1 : 2^2 : 3^2 : 4^2 \ldots$ The first or lowest critical speed is found from

$$k\,l = \pi = \sqrt{\frac{W\,p^2}{E\,I\,g\,l}}\;l$$

For a circular shaft of d in. diameter, $E = 29{,}000{,}000$, $g = 386$,

$p = \frac{2\pi}{60} N_1$

$$N_1 = 2{,}232{,}510\ d^2 \sqrt{\frac{1}{W\ l^3}}$$

for inch, pound, minute units. For a shaft with its own weight only, $W = 0.28\ \frac{\pi}{4}\ d^2\ l$. Substituting gives

$$N_1 = 4{,}760{,}000\ \frac{d}{l^2}$$

35 The values of the constants F, G, H and J, inserted in Equation 10, show that the shaft rotates in a bowed condition up to the

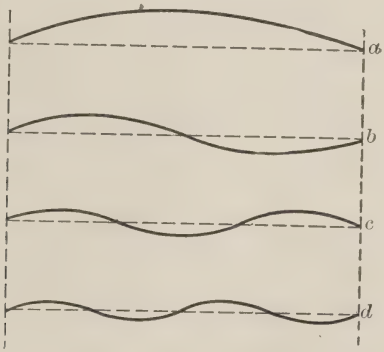


Fig. 14 Condition of Rotation at Various Speeds

first critical speed as shown in Fig. 14a; between the first and second critical speeds as in Fig. 14b; between the second and third critical speeds as in Fig. 14c; and so on.

SHAFT FIXED AT BOTH ENDS

36 A shaft with uniform load between long rigid bearings, we can treat as a beam fixed at both ends and impose the conditions upon Equations 10 and 11 of the deflection $y = 0$, when $x = 0$ and when $x = l$; and the tangent to deflection curve $\frac{dy}{dx} = 0$, when $x = 0$ and

when $x = l$. Solving these four equations for the unknown constants, each constant has the denomination of $2 - (e^{kl} + e^{-kl}) \cos kl$. y and z become infinite when the denominator equals zero, or

$$\cos kl = \frac{2}{e^{kl} + e^{-kl}} = \operatorname{sech} kl$$

37 To satisfy this equation kl is nearly $\frac{3}{2}\pi, \frac{5}{2}\pi, \frac{7}{2}\pi, \frac{9}{2}\pi\ldots$ The critical speeds have the ratio:

$$3^2 : 5^2 : 7^2 : 9^2 \quad \ldots . = 1 : 2.78 : 5.45 : 9 \ldots .$$

The first critical speed is when

$$kl = \frac{3}{2}\pi = \sqrt[4]{\frac{W\ p^2}{E\,I\,gl}}\ l$$

$$N_1 = 4{,}979{,}250\ d^2 \sqrt{\frac{1}{W\,l^3}}$$

and for a shaft with its own weight only, $N_1 = 10{,}616{,}740 \frac{d}{l^2}$ in inch, pound, minute, units.

OVERHANGING SHAFT FIXED AT ONE END

38 Taking the origin of the coördinate system at the support, we can impose upon Equations 10 and 11 the conditions of a cantilever beam; that is, the deflection $y = 0$ for $x = 0$; tangent to elastic curve $\frac{dy}{dx} = 0$ for $x = 0$; the bending moment $\frac{d^2\,y}{dx^2} = 0$ for $x = l$; and the shear $\frac{d^3\,y}{dx^3} = 0$ for $x = l$. Solving these four equations for the unknown constants, each constant has the denominator of $2 + (e^{kl} + e^{-kl}) \cos kl$. y and z are infinite when the denominator is zero, or when

$$\cos kl = -\frac{2}{e^{kl} + e^{-kl}} = -\operatorname{sech} kl$$

39 The smallest value of kl to satisfy this equation is 1.8751. The next values are nearly $\frac{3}{2}\pi, \frac{5}{2}\pi, \frac{7}{2}\pi\ldots\ldots$ Critical speeds have

the ratio of 1 : 6.34 : 17.6 : 43.6. . . . . The first critical speed is when

$$kl = 1.8751 = \sqrt{\frac{Wp^2}{E\,I\,gl}}\;l$$

$$N_1 = 795{,}196\;d^2\sqrt{\frac{1}{W\,l^3}}$$

and for a shaft with its own weight only, $N_1 = 1{,}695{,}514\ \frac{d}{l^2}$ in inch, pound, minute, units.

SHAFT FIXED AT ONE END AND SUPPORTED AT THE OTHER

40 With the origin of the coördinate system at the fixed end of the shaft, we can place on Equations 10 and 11 the condition of deflection $y = 0$ for $x = 0$ and for $x = l$; the tangent to the elastic curve $\frac{dy}{dx} = 0$ for $x = 0$; and the moment $\frac{d^2y}{dx^2} = 0$ for $x = l$. Solving these four equations for the unknown constants, each constant has a denominator of cosh kl sin kl $-$sinh kl cos kl. which equals zero for the critical speeds; or tan kl = tanh kl

$$kl = \frac{5}{4}\pi, \frac{9}{4}\pi, \frac{13}{4}\pi,$$

The critical speeds have the ratio

$$5^2 : 9^2 : 13^2 : 17^2$$

or

$$1 : 3.24 : 6.8 : 11.6$$

The first critical speed is for

$$kl = \frac{5\pi}{4} = \sqrt[4]{\frac{W\;p^2}{EIgl}}\;l$$

$$N_, = 3{,}482{,}715\;d^2\sqrt{\frac{1}{W\,l^3}}$$

and for shaft with its own weight only

$$N_1 = 7{,}021{,}600\;\frac{d}{l^2}$$

for inch, pound, minute, units.

GENERAL OBSERVATIONS

41 All formulæ developed for critical speed, for both concentrated and distributed loads, apply to vertical shafts as well as horizontal. When the shaft is vertical the equation for the Y-axis only is affected, the value of y dropping the $(-\Delta)$, the coefficient of sin pt being unchanged. Since this coefficient determines the critical speed value, we have the same critical speed for horizontal as for vertical shafts. Although some formulæ use the static deflection Δ, this is an equivalent deflection and can be used for vertical shafts by considering them horizontal.

42 The obliquity of the loads caused by the bending of the shaft has not been considered. When the load is near the bearings, as shown in Fig. 15, the load passes from the full line to the dotted line position, and back to the full line, for each revolution. The inertia of the disc offers a resistance to this change of position; and this resistance raises

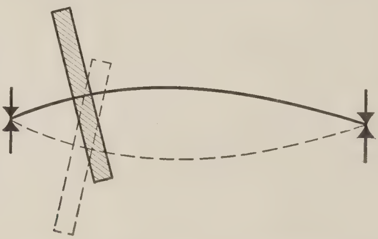


Fig. 15 Obliquity of Load during Rotation

the value of the critical speed. But the obliquity does not introduce a considerable error if the loads are nearly half way between the bearings.

43 Theoretically the vibrations become infinite at the critical speed; actually they do not, but the vibrations are at a maximum point. As shown by the curves of Fig. 3 and Fig. 8, the vibrations will begin at a certain speed, increase as the speed increases, and with still increasing speed will after a while die away. The vibrations may be felt over a considerable range, and the exact point of maximum value is difficult to detect. It is therefore advisable to keep the running speed at least 20 per cent away from the critical value; and if the normal speed is between two critical values, as in Fig. 8, careful calculations should be made for the point of minimum vibration.

44 Under ordinary circumstances the speed should be considerably below the critical value, as then the balance need not be particu-

larly good. When the speed is considerably above the critical value, the vibration is almost proportional to the unbalance (a in the equations) and the flexibility of the shaft; and the balance should be good, to prevent injury to the shaft and excessive vibration when passing through the critical speed.

45 A machine may be run very close to or at the critical speed, but the alignment and play of bearings, all mechanical details and the balance will require extra care, so that a troublesome and more expensive machine results before it is in good operating condition. The machine will run smoothly for a considerable time, until some mechanical fit or play cause a slight unbalance and immediately sets up excessive vibrations.

46 All of the solutions of shaft deflection in this paper are in the mathematical form of a harmonic vibration produced by the impressed vibration of the unbalance of load. Harmonic vibrations of this form have a special solution by calculus when p^2 equals the natural period of vibration of the shaft, or in all the cases considered, the critical-speed period. For Equations 3 and 4, when $\frac{g}{\Delta} = p^2$, the special solutions are

$$y = \frac{a\,t}{2\sqrt{\frac{g}{\Delta}}} \sin \sqrt{\frac{g}{\Delta}}\,t$$

$$z = \frac{a\,t}{2\sqrt{\frac{g}{\Delta}}} \cos \sqrt{\frac{g}{\Delta}}\,t$$

These equations show that during the critical-speed period the vibrations increase theoretically with the time, so that in machines running above the critical speed there is less vibration at the critical-speed point when it is rapidly passed over. The equations also show a transfer of energy; the kinetic energy from the unbalance being transformed into the potential energy of the shaft deflection, so that a machine with *nearly perfect* balance may run smoothly for considerable time at the critical speed before vibrations appear. The writer has not seen or had sufficient proof of the action of these two equations, but they may explain some of the peculiar phenomena observed in the vibration of certain machines.

47 With excessive vibration in passing through the critical speed

there is a considerable tendency to spring the shaft by giving it a permanent set. This is most dangerous when the machine is first started, before it has a running balance. Partly for this reason many designers use the more expensive nickel-steel forged shaft instead of carbon steel. With due consideration of the smaller coefficient of expansion of nickel-steel, in distorting large shafts when all parts are not at the same temperature, and of the fatigue or reversal of fibre stress in horizontal shafts, the machine and shaft can be so proportioned for smooth running that the finer grade of shaft steel is not always necessary.

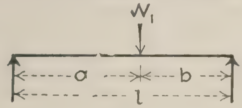
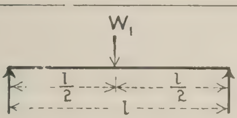
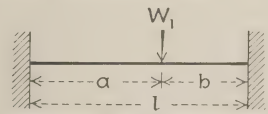
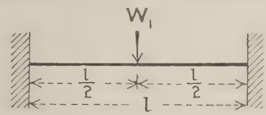
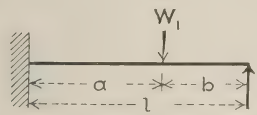
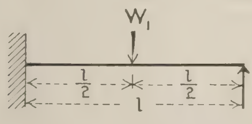
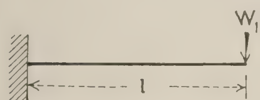
TABLE OF FORMULÆ

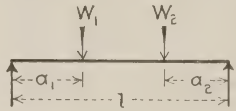
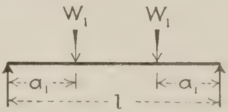
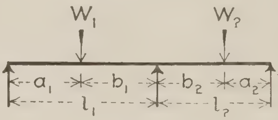
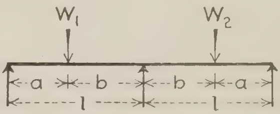
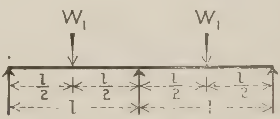
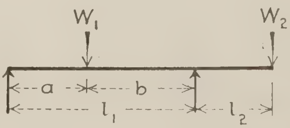
48 The formulæ developed in this paper have been transformed to suit a number of special conditions, and placed in tabulated form for convenient use. The data required for the solution of critical-speed problems are the same as those for shaft deflection at loads. As the shaft is usually of variable diameter, and its stiffness is increased by a long hub, an ideal shaft of uniform diameter and equal stiffness, or for the same deflection, must be assumed. The loads are usually concentrated with an ideal point of application. The weights and distances between bearings and loads are the same in the ideal as in the actual case. Experience has shown that when the largest shaft diameter and uniform load cover about one-third of the span, approximately the same deflection is given for the load concentrated with a uniform shaft of the largest diameter. The weight of the shaft can be divided among the concentrated loads. As formulæ have not been developed for more than two loads, when more than two loads are given they must be transformed into two resultant loads that would give the same deflection. For this case, two critical speeds are found, one of which is usually far from the working speed.

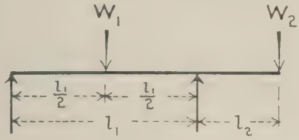
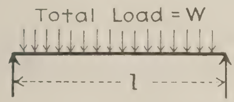
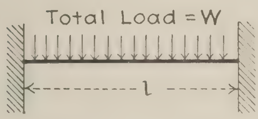
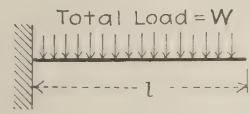
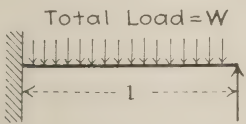
CRITICAL SPEED FORMULAE

WEIGHTS IN POUNDS, DIMENSIONS IN INCHES, VERTICAL SHAFTS CONSIDERED HORIZONTAL

N, N_1, N_2 = critical speeds in r.p.m.
Δ_1, Δ_2 = static deflections at W_1 and W_2 (shaft horizontal).
d = diameter of shaft (inches). E = 29,000,000.

Single Concentrated Load General Formulae	$N_1 = \frac{187.7}{\sqrt{\Delta_1}}$	1
	$N_1 = 387{,}000\frac{d^2}{ab}\sqrt{\frac{l}{W_1}}$	2
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 a^2 b^2}{3EIl}$	3
	$N_1 = 1{,}550{,}500\, d^2\sqrt{\frac{1}{W_1 l^3}}$	4
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 l^3}{48EI}$	5
	$N_1 = 387{,}000\, d^2\sqrt{\frac{l^3}{W_1 a^3 b^3}}$	6
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 a^3 b^3}{3EIl^3}$	7
	$N_1 = 3{,}100{,}850\, d^2\sqrt{\frac{1}{W_1 l^3}}$	8
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 l^3}{192EI}$	9
	$N_1 = 775{,}200\frac{d^2 l}{ab}\sqrt{\frac{l}{W_1 a(3l+b)}}$	10
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 a^3 b^3}{12EIl^3}(3l+b)$	11
	$N_1 = 2{,}337{,}000\frac{d^2}{l}\sqrt{\frac{1}{W_1 l}}$	12
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{7}{768}\frac{W_1 l^3}{EI}$	13
	$N_1 = 387{,}000\, d^2\sqrt{\frac{1}{W_1 l^3}}$	14
	$N_1 = 187.7\sqrt{\frac{1}{\Delta_1}}$	1
	$\Delta_1 = \frac{W_1 l^3}{3EI}$	15

Two Concentrated Loads General Formulae	$N_1 \text{ and } N_2 = 132.3\sqrt{\left(\frac{K_1}{W_1}+\frac{K_2}{W_2}\right) \pm \sqrt{\left(\frac{K_1}{W_1}-\frac{K_2}{W_2}\right)^2+\frac{4K_3^2}{W_1W_2}}}$	16
	$C=\frac{6EI}{(l-a_1-a_2)^2[l(3l-2a_1-2a_2)-(a_2-a_1)^2]}$	17
	$K_1=C\frac{2}{a_1^2}l(l-a_2)^2$	18
	$K_2=C\frac{2}{a_2^2}l(l-a_1)^2$	19
	$K_3=C\frac{l(l^2-a_2^2-a_1^2)-a_1a_2(a_1-a_2)}{a_1a_2}$	20
	N_1 and N_2 = Substitute in Equation 16	
	$N_1=548{,}400\frac{d^2}{a_1(l-2a_1)}\sqrt{\frac{l}{W_1}}$	22
	$N_2=548{,}400\frac{d^2}{a_1}\sqrt{\frac{l}{W_1(3l-4a_1)}}$	23
	$N_2=187.7\sqrt{\frac{1}{\Delta_1}}$	24
	$\Delta_1=\frac{W_1a_1^2}{6EI}(3l-4a_1)$	25
	$C=\frac{3EI}{4l_1l_2(l_1+l_2)-l_1(l_2+a_2)^2-l_2(l_1+a_1)^2}$	26
	$K_1=\left(\frac{l_1}{a_1b_1}\right)^2C[4l_2(l_1+l_2)-(l_2+a_2)^2]$	27
	$K_2=\left(\frac{l_2}{a_2b_2}\right)^2C[4l_1(l_1+l_2)-(l_1+a_1)^2]$	28
	$K_3=\frac{l_1l_2}{a_1b_1a_2b_2}C(l_1+a_1)(l_2+a_1)$	29
	N_1 and N_2 = Substitute in Equation 16	
	$C=\frac{3EIl}{2a^2b^2[4l^2-(l+a)^2]}$	30
	$K_1=K_2=C[8l^2-(l+a)^2]$	31
	$K_3=C(l+a)^2$	32
	N_1 and N_2 = Substitute in Equation 16	
	$N_1=1{,}547{,}000\,d^2\sqrt{\frac{1}{W_1l^3}}$	33
	$N_1=124.3\sqrt{\frac{1}{\Delta_1}}$	34
	$N_2=2{,}337{,}000\,d^2\sqrt{\frac{1}{W_1l^3}}$	35
	$N_2=187.7\sqrt{\frac{1}{\Delta_1}}$	36
	$\Delta_1=\frac{7}{768}\frac{Wl^3}{EI}$	37
	$C=\frac{12EI}{a^2l_2^2[4b^2l_1(l_1+l_2)-(l_1^2-a^2)^2]}$	38
	$K_1=Cl_1^2l_2^2(l_1+l_2)$	39
	$K_2=Ca^2b^2l_1$	40
	$K_3=\frac{C}{2}al_1l_2(l_1^2-a^2)$	41
	N_1 and N_2 = Substitute in Equation 16	
	$\Delta_1=\frac{W_1a^2b^2}{3EIl_1}-W_2\frac{al_2}{6EIl_1}(l_1^2-a^2)$	42
	$\Delta_2=\frac{W_2l_2^2}{3EI}(l_1+l_2)-W_1\frac{a\,l_2}{6EIl_1}(l_1^2-a^2)$	43

	$C = \frac{3EI}{l_1^3 l_2^2(l_1+l_2) - \frac{9}{16} l_1^4 l_2^2}$	44
	$K_1 = 16Cl_2^2(l_1+l_2)$	45
	$K_2 = Cl_1^3$	46
	$K_3 = 3Cl_1^2 l_2$	47
	N_1 and N_2 = Substitute in Equation 16	
	$\Delta_1 = \frac{W_1 l_1^3}{48EI} - \frac{W_2 l_2 l_1^2}{16EI}$	48
	$\Delta_2 = \frac{W_2 l_2(l_1+l_2)}{3EI} - \frac{W_1 l_2 l_1^2}{16EI}$	49
Distributed Loads.— Δ = Maximum Static Deflection.		
	$N_1 = 2{,}232{,}510\, d^2\sqrt{\frac{1}{Wl^3}}$	50
	$N_1 = 211.4\sqrt{\frac{1}{\Delta}}$	51
	$N_1 = 4{,}760{,}000\frac{d}{l^2}$ (Shaft alone)	52
	$N = [1, 4, 9, 16, \text{Atc.}]N_1$	53
	$\Delta = \frac{5}{384}\frac{Wl^3}{EI}$	54
	$N_1 = 4{,}979{,}250\, d^2\sqrt{\frac{1}{Wl^3}}$	55
	$N_1 = 245\sqrt{\frac{1}{\Delta}}$	56
	$N_1 = 10{,}616{,}740\frac{d}{l^2}$ (Shaft alone)	57
	$N = [1, 2.78, 5.45, 9, \text{Atc.}]N_1$	58
	$\Delta = \frac{Wl^3}{384EI}$	59
	$N_1 = 795{,}196\, d^2\sqrt{\frac{1}{Wl^3}}$	60
	$N_1 = 167.6\sqrt{\frac{1}{\Delta}}$	61
	$N_1 = 1{,}695{,}514\frac{d}{l^2}$ (Shaft alone)	62
	$N = [1, 6.34, 17.6, 43.6, \text{Atc.}]N_1$	63
	$\Delta = \frac{Wl^3}{8EI}$	64
	$N_1 = 3{,}482{,}715\, d^2\sqrt{\frac{1}{Wl^3}}$	65
	$N_1 = 209.7\sqrt{\frac{1}{\Delta}}$	66
	$N_1 = 7{,}021{,}600\frac{d}{l^2}$ (Shaft alone)	67
	$N = [1, 3.24, 6.8, 11.6, \text{Atc.}]N_1$	68
	$\Delta = \frac{Wl^3}{185EI}$	69

THE TRAINING OF MEN—A NECESSARY PART OF THE MODERN FACTORY SYSTEM

By Magnus W. Alexander, Published in The Journal for January 1910

ABSTRACT OF PAPER

The paper outlines briefly the educational policy of the General Electric Company at Lynn, Mass., where a systematic training is provided, suitable to all classes of people. The unskilled worker without particular education receives a training adequate to his immediate needs; the grammar school boy is initiated into the trades on the basis of a four years' course with educational instruction of a high school character; the high school graduate is trained for semi-professional service of a technical or business nature, on the basis of a three years' course with educational instruction of collegiate grade; and the college graduate is prepared for professional service of the highest order, on the basis of a two years' training of the character of a post-graduate course. An hour and a half to two hours must be spent in the classrooms every day, except Saturday, and except during parts of July and August, and classes meet during regular working hours, the students receiving the same compensation as during working hours. The apprentice training room is a trade school in the factory and permits of training under the most favorable conditions and expert supervision.

DISCUSSION

Prof. Ira N. Hollis said that this movement, which practically began ten or fifteen years ago, is bound to extend throughout the country. He referred to the city of Geneva, one of the best governed in the world, where the city council directs these courses of training, and said that the future would show whether our cities could undertake this. In the meantime these great companies are stepping into the gap.

2 The college graduate, as Mr. Alexander has pointed out, lacks practical knowledge. A committee appointed some years ago to represent all the engineering societies of England, recommended that every engineer spend at least one year in a commercial establishment before graduation, this year to follow the first year of

This paper was presented at Boston, March 11, 1910. The discussion is given in abstract only.

college work. Although this seems a prolongation, it is rather a shortening of the course, because of the value of the commercial contact. It might be difficult to get the larger firms to take men under such conditions, but it is probable that such firms would realize that they would profit very largely by the return of men thus trained into business, through their knowledge of the firm's products. Every industry and every side of our industrial life ought to have a school in an establishment where the product is made, not only for the workmen but for the graduates who are going to become engineers in that specialty.

H. S. Knowlton.[1] The industrial organization needs educational advantages among its workers. The public service corporation is coming to the same position, and within the last two or three years quite a number of companies working along street railway lines, or in central station operation, have taken this question up with considerable interest.

2 To speak specifically, perhaps one of the most interesting examples in this vicinity is afforded by the Boston Elevated Railway Company. About four years ago the company began a series of car-house foreman meetings, held monthly under the supervision of John Lindel, the superintendent of rolling stock and shops. At these meetings the defects of the cars are discussed, diagrams and charts are put on the blackboard, and the improvements in keeping rolling stock free from troubles from one year's end to another shown graphically before the men. After full discussion, the meeting is usually turned over to some representative of the operating department especially familiar with one line or another of the work, who either reads a paper or gives an informal address on some subject of special interest. Of the subjects discussed, three or four that may be cited are snow-plow maintenance, accidents, fire protection, gears and motor troubles.

3 The frank and free discussion of these topics in an intensely active operating company is bound to increase the efficiency of minor subordinate officials. Last October the Boston Elevated Railway Company started another series of meetings for its pit-men and car-house foremen and some selected shop employees. The object of the meetings held weekly has been to increase the theoretical knowledge of the work, in the belief that some classes of employees need

[1] Technical Journalist.

this, while the college graduate perhaps needs an increase in practical knowledge. Subjects that have been taken up in these lectures are static electricity, magnetism, measurement of currents, induction currents, electrical considerations in the design of direct-current motors, rheostatic control, multiple-unit control, air-brake equipment, control equipment, track maintenance, blue-print reading-records to maintain personal efficiency, and the design of various electric motors used on the system for car propulsion. Furthermore, in a shop where the work is so specialized that the foreman can follow the work of each man when piece work is undertaken, he can place the man on his own feet and assist him to make the most of his time and to cut out lost movements and other waste in the handling of his tools.

Henry Eckford Rhodes[1] spoke on the general subject of training men for factory work and referred to the methods of early apprenticeship days and his own hardships in securing a mechanical education. He pointed out the greater opportunities afforded at that time because of the greater thoroughness of the boy's training, a thoroughness which the present methods of specialization render impossible. He thought this subject to be of the greatest importance, since lack of efficient men is blocking the growth of American industries, while the non-trade branches including the professions and business occupations, are over-crowded, a condition due to the restrictions which prevent boys from securing apprenticeships in trades. He advocated the establishment of free or partly free industrial schools such as are being successfully operated in Europe, which tend to supplement the education of an ordinary school with training calculated to make the boy a more useful member of society and a larger contributor to the nation's wealth. The colleges and universities and the leading high schools in cities should be equipped with complete mechanical plants and able instructors and the electives should be extended so that greater opportunities might be given for learning trades. The work at the Naval Academy and in our technical colleges proves that it is not impossible to acquire an education and a trade at the same time and he believed that as a result of such a method of procedure many would take up a trade who are now prevented from doing so by present conditions.

[1] Passed Assistant Engineer, U.S.N., Ret.

Prof. Charles F. Park[1] spoke of the work of the Lowell Institute for Industrial Foremen, an evening school maintained by President Lowell under the auspices of the Massachusetts Institute of Technology, described in the discussion of a paper on College and Apprentice Training by J. P. Jackson (Transactions, Vol. 29, p. 507). This school for foremen was not planned for the great mass of so-called working people, but for the minority who are not uneducated, but who have been unable to gain a technical education. It comprises two courses, mechanical and electrical, each extending over two years, and including lectures, recitations, drawing-room exercises and laboratory practice. The courses are conducted by members of the faculty of the Massachusetts Institute of Technology, which has also given the use of its laboratories and class rooms. The students are required to spend two hours at the school three or four evenings a week and as many more hours in home study. There have been about ninety in the first-year class and sixty in the second class.

2 One hundred and eighty-three men have graduated. That the school is making the men more efficient in their regular occupations, and qualifying them for advancement along the lines in which they are working, has been demonstrated by these graduates. This is a strong endorsement of Mr. Alexander's statement that "training will increase their economic value and contentment, and add materially to the productive efficiency of the factory."

3 The paper presents a scheme for training men which in many ways seems ideal. To what extent this may be practically realized under our factory conditions is, in the writer's opinion, open to some question; for industrial activity, competition and numerous manufacturing considerations, the governing conditions for any plan of shop training, would necessarily influence the educational value of the work. The interest of the superintendent and foremen is in the efficiency and output of their departments; questions of instruction are secondary, and under the present scarcity of capable men, it would probably be difficult to find good foremen who would be good teachers. The students themselves, as the author says, must "never forget that they must earn as well as learn in the service of their employer." These conditions seem to be confronting difficulties in the way of thorough training from the educational standpoint.

4 This is not a criticism of shop courses as a whole, for the writer

[1] Director, Lowell School for Industrial Foremen.

believes certain parts of the man's training can be gained as well, if not better, in the factory than in the college or regular school. No course of study can possibly produce a mechanic and engineer, or an electrician, so well fitted that there is nothing left for him to gain from practical experience. His grasp of practical detail and well-balanced judgment will come only after years of service in the factory. But general training in fundamental principles of mathematics and pure applied sciences can best be given in schools which have this training for their aim and not as a secondary purpose. The mechanical industries should not be called upon to give training beyond that directly associated with their processes and methods. These industries are not philanthropic institutions and they cannot be expected to furnish such general training to our young men as we should demand from our public schools and colleges.

George Clinton Ewing[1] said the Westinghouse Electric and Manufacturing Company had a course very similar to that in the General Electric Company described by Mr. Alexander. Delegates were being sent to the various colleges, outlining the company's plan and asking the students to come to them. There are lectures in the evening rather than the day time with various topics for discussion. The students publish the Electric Club Journal, which costs more than is realized from its sale, the deficit being made up by the company. The engineers in the factory give talks and provide papers for the meetings.

Prof. E. F. Miller said that some of the Westinghouse delegates had come to the Massachusetts Institute of Technology and the prospect offered the students seemed a good one. By the company's new arrangement the men are shifted every two months, so that in the course of a year or so they have spent practically two months in every department. The students are to be put in charge of the small testing units up to 500 kw., with a more experienced man over them, and will be called upon to fill vacancies higher up in the company, as soon as these occur.

R. H. Smith.[2] The claims of the scheme outlined by Mr. Alexander are broad, and if true will cause a complete reversal of present

[1] Westinghouse Electric & Mfg. Co., Boston.
[2] Massachusetts Institute of Technology.

methods of education. If mechanical trades and technical subjects such as drawing, mechanism, machine design, pattern-making, machine construction, foundry work, steam, electrical and mechanical engineering, can be taught effectually and rapidly in the shop and factory, then schools and colleges are unnecessary and a needless expense, and our leaders in education have been near-sighted and unwise.

2 For hundreds of years the shop, factory and office provided the only opportunities for a boy to learn a trade or acquire a profession. But under the apprenticeship system the process of learning a trade was slow, illogical, and unsatisfactory to both boy and employer. There was no course, instruction, method or system, to the boy's advancement. The amount of skill and knowledge the boy acquired during his apprenticeship depended upon his own thought and calculation and upon accident, as the teaching results of the shop are small. It was pre-supposed that he would acquire information and skill by observing what went on about him, and his attempts to learn were ventures often disastrous and discouraging.

3 The factory system, with its division of labor, its countless duplication of like parts by means of highly perfected special machinery, where the operations are largely repetitional, was the means of the gradual extinction of the old apprenticeship system, and of the supplanting of the long-trained skilful mechanic by the green hand who can in one or two months be broken in to become an operator or machine specialist, and by repetition becomes expert, but with any change of work or machine is helpless.

4 How does the new apprenticeship system differ from the old? In the old system the boy was under the direction of the foreman, assisted by the journeyman of the department; in the new system he is under the direction of an instructor, assisted by an apprentice. In the old system the boy had quite a variety of work selected from a small shop. In the new system the work is selected from a large manufacturing plant and must be to a great extent work of duplition and repetition, a dulling influence on the boy's mind at that formative age when it is imperative to avoid such influences. Neither the old nor the new systems offer any organized course or systematic instruction. The boy in either case, in large measure, must rediscover elementary facts, as would not be permitted in branches of education other than in the mechanical trades. Under the new system the boy gets a few hours a week of book-learning, but pays dearly for it by having his course unnecessarily lengthened and by

doing more repetitional work, as no company could afford to give him this instruction were it otherwise.

5 Because our forefathers came to the conclusion that learning a trade was slow, illogical, unsystematic, and wasteful of time and money, schools were established and courses organized to supply the deficiencies of shop and factory, and very few would think of returning to the old system.

6 In the evolution of the many systems of teaching, the laboratory or problem method has been the most successful. This is the method in use in our engineering and medical schools, our colleges and universities, and no system in the world has trained so many men so well. It has been employed for more than thirty years at the Massachusetts Institute of Technology. A method is worked out by graded lessons from the simple to the complex, so that with the least expenditure of time a sound, systematic acquirement of information and skill is provided. In the lecture room the instructor, pre-supposing the ignorance of a class of students, begins at the foundation, thus developing the mechanical judgment of the students by making, in advance of practice, a careful study of the problem and its solution.

7 In the laboratories all students have equal opportunities to practice and apply the instruction, not as a venture, but with a clear knowledge of the method of solving, obtaining results many times more effective than can be taught in the shop or factory through constructive and repetitional work. The laboratory method creates enthusiasm and class spirit, tremendous factors in acquiring information and skill, and trains students to think logically and plan intelligently.

8 Mr. Alexander's paper says: "The very fact of this work being a part of the commercial output of the factory automatically insures a high standard of quality and quantity, and eliminates the false notions of these values usually found in purely educational trade schools." When I first came from manufacturing into the profession of teaching, I had this same idea. I thought that schools teaching the mechanical and industrial arts created false notions of accuracy, quality and quantity in the minds of the students. I thought the apprenticeship system of the firm employing thousands of men with which I was formerly connected, gave their apprentices training that could not be improved. Time and experience, however, have proved that it was I who had the false notions and ideas, and that the laboratory or problem method for teaching the principles

of the mechanical and industrial arts as a part of an engineering training, or the essentials of a trade, can never be equalled for rapidity by the apprenticeship system, however perfect, or well or generously managed. The shop and factory have not the facilities for a graded course, nor the necessary teaching knowledge. They cannot properly teach beginnings, while the school cannot teach experience. Each has a field of its own. No word is more misleading in industrial education than the word practical. The laboratory or problem method is the most practical kind of work for the learner and the only true solution of the problem of teaching the mechanical trade rapidly, economically and progressively.

9 This apprenticeship system is undoubtedly well managed and produces as good results as may be obtained or expected from any large shop or modern factory system. Mr. Alexander says: "No course has been laid out for practical work, each apprentice being advanced as fast as is consistent with his individual capacity." While there is no course and, consequently, no systematic instruction, still this system has a field of its own in training its own operators and machine specialists. The essentials of a trade may be taught also by this system provided the apprenticeship course is long enough, say, five to seven years. This will enable the boy to obtain sufficient information and skill to meet changing industrial conditions.

10 While this apprenticeship system will undoubtedly attract many boys who have neither the opportunity nor the means to learn a trade by more rapid methods, it will not find a permanent place in a modern system of education. The American people are striving for the ideal in the field of industrial education and many schemes and plans are being tried out with varying success, and I predict that when this industrial education question is settled it will be along the lines of laboratory or problem methods. This whole subject of industrial education is largely one of the repetition of the factory, and the variety of the school. Repetition is the death of ambition, advancement and development; variety is the life of energy, enthusiasm and progress.

Dickerson G. Baker. Many students of industrial conditions believe that the producing capacity of this country may be limited shortly by the number of men available for properly supervising work. This shortage of competent men is far more noticeable and serious among foremen than among higher executives, and, though rarely admitted, the efficiency of a manufacturin6 plant depends

as much upon a high average ability of foremen and gang bosses as upon executive capacity.

2 In the plant with which I am connected, there have been for years apprentice courses, averaging three years in length, for pattern makers, molders and machinists, and we have a one year course in commercial and designing engineering open to technical graduates only. But we have recently instituted and are giving our principal attention to a class of young men who have been carefully selected as having the proper qualifications to become gang bosses, rate-setters, equipment designers and foremen. Candidates must have had two or more years' machine shop practice and have given evidence of energy, thoroughness and a capacity for analysis.

3 Each student, in this course is first assigned to some machine tool, which he is taught to operate properly, and is given written instructions and a form upon which he records detailed time studies of his work, made along lines developed by Fred. W. Taylor, Past-President of the Society. The importance of realizing the full efficiency of the tool upon the class of work being done is impressed upon the student. Records of time taken for the various steps in the operation being performed, are examined daily by the foreman in charge of the class, and opportunities for improving work and saving time are pointed out. In this way the young man rapidly acquires an appreciation of the value of seconds and of the close study of details.

4 As a student becomes proficient with the operation of one tool he is assigned to others under the same conditions. After a reasonable length of time, he is permitted to make time studies of the work of other operators, and from the records of these studies we are able to make improvements in methods and equipment for the operations observed, and to establish fair and equitable piece-rates, not only reducing our manufacturing cost, but enabling our workmen to earn a considerable increase over their normal day rate. We insist upon fair and straightforward methods in dealing with the workmen in regard to these rates, and there has been practically no trouble in applying them as the student is nearly always able to demonstrate personally that the work can be done in the time assigned.

5 It is noticed that foremen trained in this way tend to proceed in an orderly manner in laying out new work in their departments, When informed as to the number of given parts to be made, they are able to determine the relation which should exist between the amount to be spent for special equipment and the amount to be spent for direct labor on the job.

6 The desirability of the student's having one year of practical experience under actual commercial conditions, between the first and second years of his technical course, brought out by Professor Hollis, is certainly one of the greatest truths before us. I should say further that if we could alternate one year of technical instruction with one year of practical experience, the results would be even better, since repetitive experience has certain great advantages absolutely necessary to the perfection of the competent executive of a manufacturing organization.

Prof. Gardner C. Anthony. I believe the plan of the apprenticeship system to be an excellent one, and a step in the right direction. My criticism relates to the amount and character of the theory to be taught in the proposed course. Problems will probably arise similar to those with which the engineering schools have been struggling in adapting the several courses in shopwork to the curriculum, and we find today a great variety of methods employed for giving this instruction.

2 All classes of shop work should be taught for their educational value, rather than for the information which may be acquired through them. If a course in machine tools for example, is not closely articulated with such subjects as physics, mechanics, mechanism, design, etc., and subjected to the same tests for educational development that are given in other courses, this work had better be relegated to the factory. But because such courses can be made of great pedagogical value by properly articulating them with other courses, they have a proper place in the curriculum of the engineering school.

3 The new form of education under discussion may find a like difficulty in incorporating such subjects as analytical and descriptive geometry, elementary calculus, thermodynamics, etc., into the new curriculum of the factory. This is more likely to occur if an attempt is made to duplicate the courses now given in the colleges, using similar text books and conducting the work in the same manner.

4 A course in descriptive geometry might be given which would not occupy more than one-half the time devoted to that subject in engineering schools; and it should be of a different character more closely allied to the problems to be met in the shop, which necessitates a special text book prepared by those in charge of such a course. This same suggestion would apply to analytical geometry. It will require considerable skill on the part of the instructor to make the element of calculus a live topic in the midst of the pressure of the more practical subjects, although I believe that it can be done.

5 If the subjects can be thus closely related, I believe that the new form of training will be well adapted to a large number of young men who will become capable of filling the better class of positions in engineering establishments.

Prof. Peter Schwamb thought the separation of the instruction department from the shop a wise one, since it brings the student under better supervision and his advance can be made more rapid, if not too much repetitional work is required.

2 While the treatment of the students as individuals in their manual work may advance the better ones more rapidly, the usual effect will be that they obtain more of the instructor's time than is their due, since he will naturally want to instruct where his efforts will make the best showing. The adoption of the class method of instruction will beget a friendly spirit of rivalry among the students and save much valuable time of both instructors and students.

3 The advance of the class should be as rapid as is consistent with the production of good work and the thorough mastering of the operation, the element of expertness being left for future training. The student may be made familiar with the fundamentals, his class-room training being properly correlated to his shop work. After such fundamental training the student may be safely started upon commercial work, with a view of obtaining expertness on the various tools.

4 Mr. Alexander's plan can probably be improved by the adoption of definite courses systematically arranged for the beginners in all departments, such courses covering the fundamental principles and operations in logical order, and it is suggested that such a plan be introduced in the foundry instruction department yet to be equipped. Such courses would also offer a greater attraction to technical graduates, who could obtain through them a much desired practical experience and be brought into close touch with commercial engineering work.

Luther D. Burlingame said there is no substitute for patient and continued practice at a trade and that the apprenticeship system is the most satisfactory means of producing skilled workmen. Its efficiency is greatly increased however by combining with it auxiliary training to supplement the main line of work, such as mathematics, drafting, mechanics, etc., for the machinist apprentice, and varied machine shop work for the draftsman apprentice, these illustrations being typical of the needs in all trades.

2 The value of the shop school is that it makes auxiliary training a necessary part of the course of apprenticeship for which the apprentice is paid and during which he is under full control of his employers. This can be acquired in an evening school, but unless compulsory a large number of those needing this training most would not avail themselves of it.

3 There is a wide field for the employment of different methods in carrying out a system of industrial education in the shop. In the Brown & Sharpe works there are about 150 apprentices coming under some form of auxiliary training. The machinists' apprentices, instead of working part of their time in a machine shop apprentice training room, spend all of their four years of apprenticeship, except for school work, in the regular shop departments. This plan does not require the duplication of machines in another department and it keeps the boys in touch with the more advanced machine operations, so that they can gain experience constantly by observing work going on about them as well as from what they are doing themselves. It brings about contact with the various foremen during the entire period and this mutual acquaintance is helpful in determining the boy's value and, from the apprentice's standpoint, in getting into the spirit of the shop and its personnel. During the time these apprentices are working in the machine shop they are given work in eight or more of the regular departments, thus becoming acquainted with foremen and workmen as well as with methods. No shop should be deterred from establishing an apprenticeship system because it does not feel justified in equipping an apprentice training department where a separate equipment of machines is required.

4 In the Brown & Sharpe works, without the use of text books and without the learning of rules, etc., the problems are presented as they would arise in the shop, except that they are in regular sequence as to subject and difficulty. They are taken up with such reference tables and books at hand as should be in the possession of intelligent mechanics, and the boys are taught how to use such means to solve the problems. They are not taught geometry, algebra, trigonometry, etc., as such, but learn quickly to apply such principles of these sciences as are needed for the problems arising in the shop, and perhaps before knowing these sciences even by name are making practical use of them in their work. Instead of learning certain rules and then applying them, the application comes first showing what the rule must be, or, at least, where in the ordinary

reference books it can be found. Bettei still, the method of solving a problem is often worked out by the boys based on what they have previously done, and making them to that extent independent both of memorized rules and reference to text books. The whole course is directed toward cultivating the reasoning powers rather than the memory, and gives a chance for the intelligent grammar school graduate to hold his own better than would be expected, in comparison with the ordinary high-school graduate of the same age. The aim is to make skilled machinists, and while this course fits also for foremanships and other lines of advancement, the greatest need of today is for skilled workmen.

5 The speaker also exhibited several blue-prints showing problems to be worked out by the students. These related to linear measurements, fractions and decimal equivalents, screw threads, tapers, gearing, etc., certain data being given from which discussions or other results are to be determined by calculation.

The Author.[1] It was never my intention nor anybody's intention that technological or other schools should be abolished. Let me refer to one or two things. Mr. Smith claims that the repetitive character of our work, so necessary, is absolutely useless in any attempt at training efficient mechanics in these shops. But does he forget that the very same jigs and fixtures that are needed in such large numbers and high efficiency must be made by intelligent mechanics? It does not matter how much repetition work is produced by these jigs. We must have men who can design these jigs and fixtures and can design proper machinery on which these jigs and fixtures can be used.

2 Also a certain amount of repetitive work is not only not bad, but is absolutely necessary for the proper training of a mechanic. The intensity of production which can be taught only through repetitional work must be applied to our growing generation. It is absolutely necessary. You can't teach the intensity of production and all that it means by having a young man make a special tool or a jig or a fixture, because you have no proper, definite measure of the time it should take. But if you let that young man make 50 motor-shafts, you can hold him down not only to absolute commercial accuracy, but to a very fair degree of speed with his own hands and with his machine. How about the many high school graduates who

[1] This discussion was not revised by the author.

cannot or do not want to go to a technological school or to a college? And how about the 75 per cent of the grammar school graduates who never go beyond the public grammar school because they cannot or do not want to enter our industries? All these people, both the high school and the grammar school boys, making almost 100 per cent, must receive mechanical training, if they receive it at all, in the shop, which is the proper place for them to receive it.

3 Should we leave out the class room work because we cannot give it as well as it is given in college? And, let us say, we do not give it as well, because we are not professional teachers. We are professional business men, trying to apply good, sound business principles to education. I think it would be a misfortune to leave it out. Our class room is not so much intended to give definite, concrete knowledge in mechanics. It has a far broader purpose, and the first purpose of all is to give these young men who are growing up and who are going to be our industrial army and our industrial non-commissioned officers, an objective as well as a subjective viewpoint. Our whole labor problem hinges on the fact that our men have only a subjective viewpoint and need an objective one, also; that they cannot see things from the other side as well, but only from their own side. In order to give them that objective viewpoint; in order to develop the character of the men; in order to make them well-intentioned men, good citizens, good working citizens, good workingmen, we have instituted, and I believe every manufacturer should institute, some class room work.

4 We must develop the intelligence at the same time that we are developing the hand. And it is not essential whether the instructor in our class room, pedagogically speaking, toes in or toes out when he is before his class.

TOPICAL DISCUSSION ON RECENT DEVELOPMENTS IN WHEEL TESTING

Dr. C. H. Benjamin

Before proceeding to describe the testing pit recently established at Purdue University for experimental work in bursting various rotating members by centrifugal force, it will perhaps be well to review the progress made in this kind of experimentation since the first work which was reported to the Society in 1898. At that time, the object was to determine the bursting speed of small model flywheels with different types of rims and joints.

2 The first wheels experimented on were 15 in. in diameter. They were rotated by means of a Dow steam turbine, the speed being measured by an electric commutator. The shield used was made of 2-in. pine plank, weighted with heavy castings and timbers. Fig. 1 shows the appearance of the shield after the first explosion. For succeeding experiments, a similar shield of 6 in. by 12 in. white oak was constructed.

3 An increase to 24 in. in the size of the wheels tested resulted in the complete wrecking of this shield, as may be seen by Fig. 2. In all further experiments conducted that year, the shield was made of oak timbers 12 in. square, firmly bolted together and covered with 3-in. oak plank. The speed of the flying fragments is indicated by the fact that some of them cut clean holes through moving belting, similar to those which would be made by bullets. During this year, ten 15-in. and seven 24-in. wheels were broken.

4 Further experiments on 24-in. model wheels were made during the following year and sixteen wheels were broken. The wheels in this series of experiments were enclosed in a cast-steel ring 36 in. in inside diameter, with a rim section 4 in. by 6 in. This was lined

This discussion was presented at St. Louis, March 12, 1910.

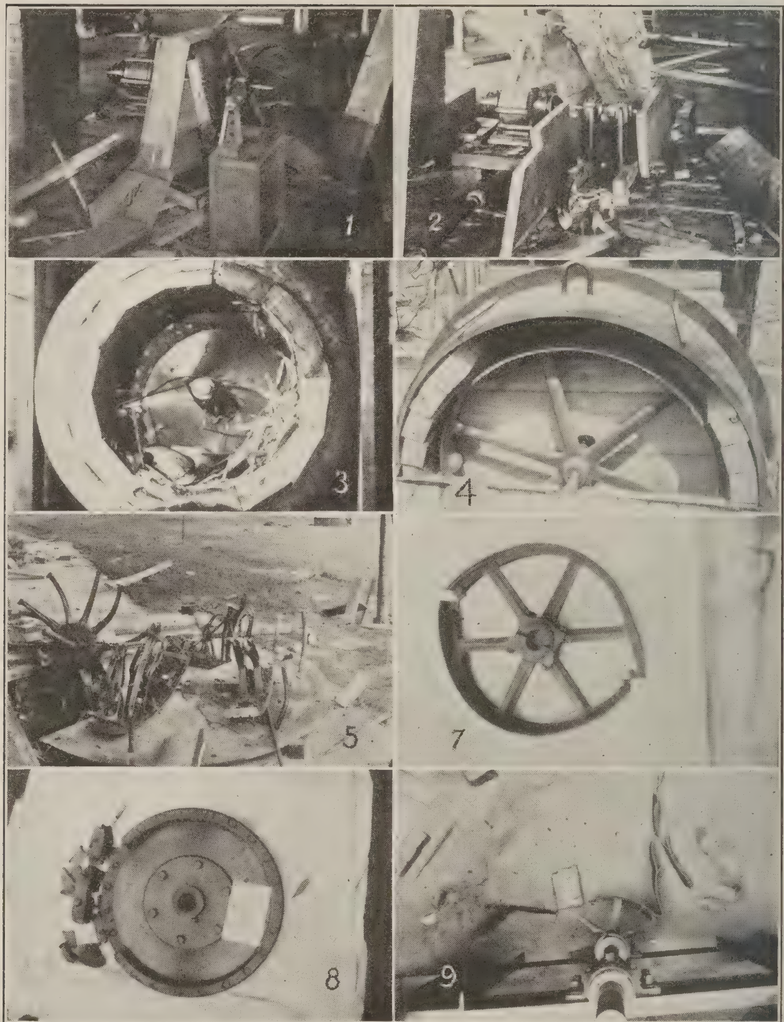


Fig. 1 Shield of 2-in. Pine After Explosion of 15-in. Wheel
Fig. 2 Shield of 6-in. by 12-in. Oak After Explosion of 24-in. Wheel
Fig. 3 Cast-Steel Ring After Explosion of 24-in. Wheel
Fig. 4 Steel Shield for Four 4-ft. Pulleys
Fig. 5 Wreck of 4-ft. Wheel and Shield
Fig. 7 24-in. Steel Pulley Showing Weak Joint
Fig. 8 Paper Pulley
Fig. 9 View Looking Down Into Pit After Explosion of a Steel Pulley

with wooden blocks to absorb the energy of the fragments, and was completely enclosed in oak planking. The same steam turbine was used for driving the pulleys, but a tachometer was used for indicating the speed.

5 The appearance of the casing and wheel after an explosion is shown in Fig. 3. After such an explosion, the wooden blocks would move around in the ring several inches, showing the tangential motion of the fragments. That this apparatus was not entirely safe was demonstrated in one experiment by the escape from the casing of portions of the pulley rim, due to the breaking of the retaining bolts. Although there were numerous spectators in the room, no accident occurred.

6 In this same apparatus fifteen emery wheels of different makes were burst, as reported to the Society in 1903. On account of the greater fragility of the emery wheels, no accident resulted.

7 These experiments were followed in the succeeding year by tests on wooden and steel pulleys 24 in. in diameter. The results of these experiments were published in August 1905 in Machinery. Eight pulleys were tested, including five wood split pulleys, one with steel arms, one all-steel pulley, and one wooden pulley with a solid web which we did not succeed in breaking. A number of pulleys made of paper fiber were tested in the same way, with results not very different from those on wooden pulleys.

8 In 1906 and 1907, a large number of cast-iron discs of various thicknesses and types of hub were exploded in the same apparatus. The results of these experiments, and the conclusions from them, have not yet been published.

9 It seemed desirable to test larger pulleys in the same manner, to see if the peripheral bursting speed would be the same for different sizes of pulleys. A few experiments were made in 1904, on pulleys 4 ft. in diameter. Former experiments had showed that it was hardly safe to burst pulleys of this weight and size inside a building. The apparatus shown in Fig. 4 was built entirely of steel and was 5 ft. in inside diameter. The shield was of rolled boiler plate, $1\frac{1}{4}$ in. thick and having a tensile strength of 65,000 lb. per sq. in. Flat plates $\frac{3}{8}$ in. thick were bolted to the sides so as to enclose completely the wheel tested. Fig. 5 shows the shield after the explosion of the third wheel. The upper half of the casing, weighing about half a ton, was carried 75 ft. in the air and some hundred feet in a horizontal direction.

10 About this time the attention of the writer was called to a vertical shaft used by a German experimenter for bursting emery wheels,

the wheel being mounted at the lower end of the shaft inside a pit. The simplicity and safety of this form of construction are strong points in its favor. The last testing apparatus devised is constructed on this principle and located at Purdue University.

11 Fig. 6, showing a vertical section of the pit and connections, needs little explanation. The weight of the wheel and shaft is supported by a ball and thrust bearing, while rotation is effected by a 10 h.p. motor having a speed which can be varied from 800 to 2400 r.p.m. The pit itself is lined with concrete, but the impact of the fragments is received by a bank of sand. This works admirably to prevent any bruising or smashing of the fragments after the explosion. The speed is taken in the usual way by a tachometer.

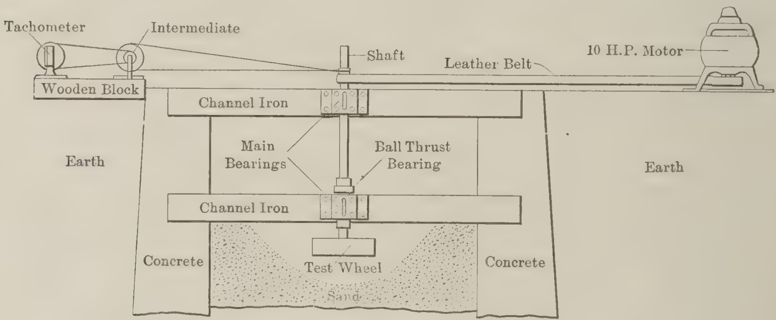


Fig. 6 Flywheel Testing Pit at Purdue University

12 During the winter of 1908–1909, this apparatus was used very successfully[1] in testing the strength of sixteen pulleys, all 24 in. in diameter, with rims from 6 in. to 6¾ in. wide. The material used in their construction was wood, cast iron, paper and steel. Some of the rims were solid but most of them were of the usual split pulley type. The linear bursting speed of the solid wooden pulleys was about 275 ft. per sec., or 2600 r.p.m. The linear bursting speed of the split pulleys varied from 220 to 260 ft. per sec., or from 2100 to 2600 r.p.m. The paper pulleys on the other hand, having a solid web, were considerably stronger, averaging about 300 ft. per sec., linear bursting speed, or nearly 2900 r.p.m. Contrary to the usual opinion, the steel wheels are no stronger against bursting than the average wooden pulley. In fact, they are somewhat weaker than a well constructed pulley made of wood. Two wheels tested burst at exactly the same speed, 2240 r.p.m., or 235 ft. per sec.

[1] These experiments were made by Messrs. Biggs and Woodworth, senior students, as a part of their graduating theses.

13 The weakness of this type of pulley is due to the peculiar form of joint fastening, which is bent and broken by the centrifugal pressure. The bursting of the wooden pulleys was due in most cases to the greater density of the balance weights, consisting of slugs of round iron inserted in holes bored in the rim. These caused considerable centrifugal force at the points where they were located. It was evident from the appearance of the broken wheel that some of these weights had forced their way through the rim, thus starting rupture.

14 It is difficult to see how ordinary pulleys with wooden rims can be satisfactorily balanced without weakening. As there is rarely necessity for a linear speed of more than 100 ft. per sec., however, all of the pulleys tested had a factor of safety sufficient for commercial use. This is not true, however, of all pulleys. Two 4-ft. pulleys which were tested burst at speeds of 1100 and 600 r.p.m., respectively, which was considerably less than was expected of them. In the case of the pulley having a solid rim, this was due to the presence inside the rim of a balance weight of $3\frac{1}{2}$ lb. At 1100 r.p.m. the centrifugal force of this balance weight was over 2700 lb. In the same manner, 4-ft. pulley No. 2 was burst by the centrifugal pressure of a flange which weighed with its bolts $7\frac{1}{2}$ lb., and had a centrifugal force at bursting speed of nearly 1700 lb.

15 The effect of a joint flange is particularly disastrous, on account of the weakness of the joint itself to resist bending.

CONCLUSION

16 The bursting speed of most cast-iron pulleys having continuous rims may be put at about 400 ft. per sec., corresponding very nearly to a centrifugal tension of 16,000 lb. per sq. in. A wooden pulley with a continuous web and rim is even stronger than this, since wood is stronger in proportion to its weight than cast iron. A 2-ft. wooden pulley of this description has been run at a speed of 467 ft. per sec. without breaking. The ordinary split pulleys, whether of wood, steel or iron, cannot be relied upon at speeds much over 200 ft. per sec., on account of the weak points which have been mentioned. For experimental high speeds, steel pulleys of a much higher bursting point could undoubtedly be constructed. The poor joint design of the ordinary split steel pulley, such as is used for shafting transmission, renders it unusually weak in this respect.

17 It is proposed to use the testing pit for further experiments along several different lines, one being the testing of various kinds

of grinding wheels, including carborundum as well as the ordinary wet grindstone. The writer also hopes to test out several flywheel joints, on model wheels ranging from 4 ft. to 6 ft. in diameter; also to have some time an opportunity to test some band saw wheels in a similar manner.

DATA AND RESULTS OF EXPERIMENTS MADE IN THE NEW TESTING PIT

No. of Test	Kind of Material in Pulleys	Rim				Weight Pounds	Bursting Speed	
		Style	Diameter Inches	Breadth Inches	Depth Inches		r. p. m.	Peripheral Speed Ft. per Sec.
1	wood	solid	24	6.25	1.62	29.37	2720	284.7
2	wood	solid	24	6.25	1.62	29.37	2550	266.9
3	wood	2 sections	24	6.5	1.78	29.67	2210	231.8
4	wood	2 sections	24	6.5	1.78	29.67	2110	220.8
5	wood	2 sections	24	6.5	1.78	28.81	2390	251.0
6	wood	2 sections	24	6.5	1.78	28.81	2430	254.3
7	wood	2 sections	24	6.5	1.78	28.81	2360	247
8	wood	2 sections	24	6.5	1.78	28.81	2420	253.3
9	wood	2 sections	24	6.5	1.78	28.81	2570	258.5
10	wood	2 sections	24	6.5	1.78	28.81	2535	244.4
11	cast iron	solid	24	6.0	0.406	70.44	3720	389.4
12	cast iron	solid	24	6.0	0.406	70.44	3380	353.8
13	paper	solid	24	6.0	1.75	77.37	2820	295.2
14	paper	solid	24	6.0	1.75	77.37	2930	306.7
15	steel	2 sections	24	6.75	0.0625	41.75	2240	234.5
16	steel	2 sections	24	6.75	0.0625	41.75	2240	234.5

FURTHER DISCUSSION

Following the introductory discussion by Dr. Benjamin, the question was asked by G. M. Peek if it had been found necessary, after the bursting of a pulley, to renew the driving shaft on which it had been mounted. Dr. Benjamin replied that the shaft was usually sprung by the unbalanced rotation of the pulley after bursting, and had to be replaced by a new one.

2 J. D. McPherson submitted the accompanying sketch (Fig. 1) of a large flywheel with heavy arms split on the centre of an arm. The two parts are dovetailed together and held by prisoners. He explained that this construction avoids the objectionable practice of placing the weight of the joint between the arms.

3 Dr. Benjamin stated that in this way any desired joint efficiency, up to 80 or 90 per cent, could be obtained. There can be no bending action due to centrifugal force and therefore no tendency to open the joint. Most split flywheels are now made with some form of joint over the arms.

4 H. A. Ferguson asked whether any experiments had been made on steel discs such as those used for the cold-sawing of structural steel. They are of open-hearth flange steel and run at a peripheral speed of 22,000 to 26,000 ft. per min. They are frequently replaced b cause of splits on the periphery, but this is apparently due to crystalization caused by the alternate heating and cooling to which they are subjected.

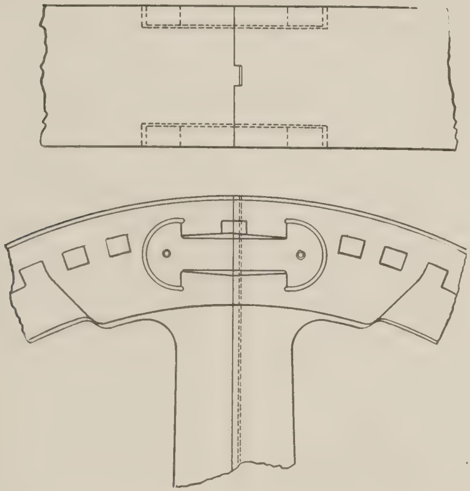


FIG. 1 LARGE FLYWHEEL WITH HEAVY ARMS SPLIT ON CENTER OF ARM

5 Dr. Benjamin said that little is known experimentally about the bursting speed of discs, but it is certain that they are stronger than rings, because the web resists the centrifugal tension. Stodola's Steam Turbines, the best theoretical work on the subject, states that the bursting strength of a ring is half that of a solid disc, but the speaker had never believed this statement, although it is theoretically correct. Solid steel discs are probably safe at the speed mentioned, but the practice of punching holes in the discs to balance them is questionable. Even a solid disc might be wrecked by centrifugal force in combination with the severe strains due to the cutting action.

6 Col. E. D. Meier suggested that, since flywheel problems deal with those of 10 or 12 ft. in diameter, and the expense of actual experiments on such wheels would be too great for private institutions, the engineering profession should use its influence to have such experiments made by the Government or at Government expense in private institutions.

7 Prof. H. Wade Hibbard announced that a series of experiments on aeroplane propellers were to be made this spring at the Missouri State University. The propellers are to be of wood, 6 ft. in diameter, and will be run at the speeds used in actual practice, i.e., 600 to 1800 r.p.m. It had been the intention to make the tests in the laboratory, but the experiences of Professor Benjamin with the bursting of flywheels would seem to indicate that this would be dangerous.

8 Dr. Benjamin responded that he would consider any such experiments extremely unsafe unless the apparatus were enclosed with some strong material. No matter what the material, when speeds of from 200 to 400 ft. per sec. are attained, it is vicious. In some of his experiments he found that fragments of wood went through a running belt, leaving holes as clean as a bullet would.

9 Colonel Meier suggested that Professor Hibbard's experiments be extended by testing some propellers to destruction in order to determine the factor of safety of those now being used.

GENERAL NOTES

AMERICAN SOCIETY OF CIVIL ENGINEERS

The annual convention of the American Society of Civil Engineers will be held in Chicago June 21–24, 1910.

At the regular monthly meeting on May 4, two papers were read: Water Supply of the El Paso Southwestern Railway from Carrizozo to Santa Rosa, New Mexico, by J. L. Campbell; and The New York Tunnel Extension of the Pennsylvania Railroad: The site of the Terminal Station, by G. C. Clarke. On May 18, J. C. Meem presented a paper entitled Pressure Resistance and Stability of Earth.

AMERICAN INSTITUTE OF MINING ENGINEERS

On Saturday evening, April 30, the American Institute of Mining Engineers gave a dinner in celebration of the seventieth birthday of Dr. Rossiter W. Raymond, for the last thirty years Secretary of the Institute. Nearly 400 were in attendance.

A gold medal from the Institution of Mining and Metallurgy was presented to Dr. Raymond by R. T. Bayliss, and illuminated parchments which were sent by various foreign engineering bodies were presented by E. G. Spillsburg, Mem. Am. Soc. M. E. Dr. Raymond was also made the recipient of a silver service at the close of the speaking.

Among those who paid tribute to Dr. Raymond were Dr. James Douglas; Dr. Lyman Abbott; George Westinghouse, President Am.Soc.M.Ee.; John Bensel; M. Sorzano de Tajada, of the Société des Ingénieurs Civils de France; Frank Dawson Adams, president of the Canadian Mining Institute; Robert W. Hunt, Past-President, Am.Soc.M.E.; Thomas Commerford Martin; William Lawrence Saunders, Mem.Am.Soc.M.E.

The guests were presented with an illustrated booklet containing scenes from the life of Dr. Raymond.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

The annual convention of the American Insitute of Electrical Engineers will be held June 27–30, at the Waumbek Hotel and cottages at Jefferson, N. H., in the White Mountains. The annual meeting was held at New York May 17. Officers were declared elected as follows: president, Dugald C. Jackson, Mem. Am.Soc.M.E.; vice-presidents, Percy H. Thomas, H. W. Buck, Morgan Brooks, Mem.Am.Soc.M.E.; managers, H. H. Barnes, Jr., C. E. Scribner, W. S. Rugg, R. G. Black; treasurer, Geo. A. Hamilton; secretary, Ralph W. Pope.

On May 5 to 7 a meeting under the auspices of the High Tension Transmission Committee, of which Ralph D. Mershon, Mem.Am.Soc.M.E., is chairman, was held at San Francisco. Elaborate arrangements were made for the entertainment of the members, and professional papers were read as follows: Emergency Generating Stations for Service in Connection with Hydroelectric Transmission Plants under Pacific Coast Conditions, A. M. Hunt, Mem.Am. Soc.M.E.; Hydroelectric Power as Applied to Irrigation, J. C. Hays; The Developed High-Tension Net-Work of a General Power System, Paul M. Downing; Parallel Operation of Three-Phase Generators with their Neutrals interconnected, G. I. Rhodes; Observations of Harmonics in Current and Potential Wave Shapes of Transformers, John J. Frank; Transmission Line Crossings of Railroad Right-of-Way, A. H. Babcock.

NATIONAL ASSOCIATION OF COTTON MANUFACTURERS

The stated annual meeting of the National Association of Cotton Manufacturers was held in the Mechanics Fair Building, Boston, Mass., April 27 and 28, 1910.

An address of welcome was made by Governor Draper of Massachusetts, to which Franklin W. Hobbs, of Boston, responded. Addresses followed by the president, Charles T. Plunkett, Mem.Am.Soc.M.E., Richard C. Maclaurin, President of the Massachusetts Institute of Technology, and Howard Ayres, secretary of the Cotton Goods Export Association. Papers were presented at the sessions of Wednesday afternoon and Thursday on The Progress of the Diesel Engine, Col. E. D. Meier, Mem.Am.Soc.M.E.; The Federal Corporation Tax Law, Walter S. Newhouse; A Substitute for Cotton, James Hope; Superheated Steam and Superheaters, Dr. D. S. Jacobus, Mem.Am.Soc.M.E.; The Electric Drive as a Manufacturing Proposition, Meldon H. Merrill; Choice of Power for Textile Mills, Charles T. Main, Mem.Am.Soc.M.E.; Recent Advances in the Chemistry of Coal Tar Colors, Dr. Hugo Schweitzer; Sizing of Vegetable Fibers, Hermann Seydel; Production-Increasing Methods, Henry L. Gantt, Mem. Am.Soc.M.E.; Distribution of Artificial Light, F. M. Scantlebury; Bibliography of the Cotton Manufacture, Dr. C. J. H. Woodbury, Mem.Am.Soc.M.E.

The following officers were elected: Franklin W. Hobbs, President; George Otis Draper and Edwin Farnham Greene, Vice-Presidents; Albert F. Bemis, R. M. Miller, Jr., Russell B. Lowe, Frederick A. Flather, Mem. Am. Soc. M. E., and Frederick B. Macy, Directors.

LECTURES ON AËRIAL NAVIGATION AT MCGILL UNIVERSITY

The Department of Mechanical Engineering of McGill University has arranged a course of lectures on aërial navigation which will deal with the mechanical principles involved in the construction of the machines, the process of their manufacture, the difficulties of steering and manipulating with the various methods in use to obviate these troubles, and the displacement of air and the theory of gliding. The course will be in charge of Prof. C. M. McKergow.

ADVANCED STUDY OF ELECTRICAL ENGINEERING AT MASSACHUSETTS INSTITUTE OF TECHNOLOGY

The Massachusetts Institute of Technology will this year confer for the first time in its history the degree of Doctor of Engineering. Special attention will be given next year to graduate work. The lectures of Prof. Harold Pender will extend the discussion contained in his advanced lectures of this year on the high-voltage alternating transmission and utilization of power, with a repetition of the general treatment in his lectures of this year on the transmission circuit, and more attention will be given to the conditions arising from the utilization of power. Professor Jackson's lectures for graduate students on the organization and administration of public service companies will next year be directed more to the theory underlying methods of charging for service by public service companies, with particular reference to charges for electric light and power, but with collateral consideration of railroad and tramway charges and charges for gas and the service of other public utilities. Professor Wickenden will originate a course of lectures on illumination, photometry and illuminating engineering, as a part of the optional curriculum for undergraduate and graduate students.

FOREST PRODUCTS LABORATORY AT UNIVERSITY OF WISCONSIN

The Forest Service of the United States and the University of Wisconsin are coöperating in the establishment at Madison, Wis., of a Forest Products Laboratory, which will be prepared to carry on tests of the strength and other properties of timber, the preservative treatment of timber, the saving of wood waste by means of distillation processes, and the fiber of various woods for paper and other purposes. It is proposed to make it the largest and best equipped wood-testing laboratory in the world. The laboratory will be formally opened on June 4, 1910, and representatives of lumber manufacturing and wood-using associations from all parts of the country are expected to attend.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 12th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

026 Young technical graduate with good scholastic record and at least two years practical experience, for position of assistant in the laboratory of an engineering school; salary $1000 for the academic year. Location, Massachusetts.

027 Instructor in the mechanical engineering department of Columbia University, New York; will pay $1000 per year.

028 Manager of well-known company in New York State requires the services of an active, capable, educated and energetic man of good address, tactful in the management of men, familiar with approved systems of commercial and workshop management and costs. Candidate for such position should have thorough practical experience in machine shop and foundry producing heavy machinery for steel works, mills, etc.; large steam and gas engines, general heavy jobbing work and gas producers, also high-class marine and stationary boilers and heavy steel-plate work.

029 Engineer, experienced in the design of hydraulic machinery, pumps and large rolling mills for rolling sheet metal; must be thoroughly competent to make estimates and prepare complete calculations and data for drafting room. Location, Connecticut.

030 Michigan concern engaged in furnishing to several companies on co-operative basis, electric power, live steam, low pressure steam for heating, gas and compressed air, wishes to engage man of executive ability to take entire charge of plant and its operation; he must be qualified to give expert advice on changes or alterations, keep up output and give satisfactory service. Wants thoroughly high-grade competent man.

MEN AVAILABLE

73 University graduate, B.S.M.E., general experience with consulting engineer; five years in engineering and executive positions; building power plant machinery. Past six years with electric railway system in charge of design and construction of modern power house equipment, large units, high and low-pressure turbines, condensers, cooling towers, steam and gas engine plants. Will change on reasonable notice. Eastern location preferred.

74 Junior member, M.E., would like to connect with some consulting engineer, in or around New York. Industrial engineering preferred.

75 Young man desirous of getting away from close application to drafting board; technical graduate; experience in office and construction work as well as drafting, on power and industrial plants, wants commercial work with firm of engineers and contractors or with industrial company.

76 Engineer, 9 years experience in civil engineering, especially hydraulics; 5 years in charge of experimental steam turbine work; desires position as assistant professor of civil or mechanical engineering.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ABERCROMBIE, James Henderson (1901), Mech. Supt., Clark Thread Co., Clark and Ogden Sts., Newark, N. J.

ALBERGER, Louis R. (1889), Pres., Alberger Condenser Co., 140 Cedar St., New York, N. Y.

ATKINS, David Fowler (1907), Supv. Architects Office, Treasury Dept., Washington, D. C.

BIBBINS, James Rowland (1904; 1909), Engr. with Bion J. Arnold, 154 Nassau St., New York, N. Y.

BILLINGS, William Richardson (1906), Secy. and Treas., Alberger Condenser Co., and Alberger Pump Co., 140 Cedar St., New York, and 151 Columbia Hgts., Brooklyn, N. Y.

BIXBY, William P. (Junior, 1908), Mech. Dept., Erie R. R. Co., 50 Church St., New York, N. Y.

CAMPBELL, Gordon M. (1906), Genl. Elec. Co., Turbine Dept., West Lynn, Mass.

CASH, Arthur Wise (1899), Charge Regulating Valve and Engrg. Dept., H. Mueller Mfg. Co., Decatur, Ill.

CHAMBERS, Norman C. (Junior, 1905), Sales Dept., Niles-Bement-Pond Co., 111 Broadway, New York, N. Y., and *for mail*, care F. H. Bagge, Calle San Martin, 121, Buenos Aires, Argentine Republic, South America.

CREELMAN, Frank (1894), 447 W. 23d St., New York, and *for mail*, Hotel Cunningham, Sandy Hill, N. Y.

DAUGHERTY, Samuel Bovard (1905), Ch. Draftsman, Gas Eng. Dept., Snow Steam Pump Wks., and *for mail*, 129 N. Norwood Ave., Buffalo, N. Y.

DECKER, Edward P. (1906), E. P. Decker & Co., 80 Griswold St., and 79 Pingree Ave., Detroit, Mich.

ELLICOTT, Edw. Beach (1903), Elec. Engr., Sanitary Dist. of Chicago, 1500 Am. Trust Bldg., and *for mail*, 6229 Winthrop Ave., Chicago, Ill.

ESTES, William Wood (1891; 1904), Designer, Genl. Fire Extinguisher Co., and *for mail*, 245 Waterman St., East Side Sta., Providence, R. I.

FRANCIS, W. H. (1884), Union League, Broad and Sansom Sts., Philadelphia, Pa.

FRITZ, Aime L. G. (Junior, 1907), Ch. Draftsman, Hartford Suspension Co., 150 Bay St., Jersey City, N. J., and *for mail*, 99 Elmwood St., Woodhaven, L. I., N. Y.

GATES, Philetus W. (1902), Vice-President, 1906-1908; Pres., Hanna Engrg. Wks., 2059 Elston Ave., Chicago, Ill.

GIBBS, Geo. (1890), Ch. Engr. Elec. Traction and Terminal Sta. Constr., Pa. Tunnel & Terminal R. R. Co., Ch. Engr. Elec. Traction, West Jersey & Seashore R. R. Co., L. I. R. R. Co., 32d St. and Seventh Ave., New York, N. Y.

HAMERSTADT, William Diehl (Junior, 1907), Engr., Rockwood Mfg. Co., and *for mail*, 1608 Central Ave., Indianapolis, Ind.

HARTNESS, R. B. (Associate, 1903), 3042 Foster St., Los Angeles, Cal.

HECK, Robert C. H. (1906), Prof. Mech. Engrg., Rutgers College, and *for mail*, 35 College Ave., New Brunswick, N. J.

HEIKEL, Daniel August (1899), Life Member; M. E., Rm. 745, Oliver Bldg., Pittsburg, Pa.

HILL, E. Rowland (1907), Asst. to Ch. Engr., Elec. Traction, Pa. Tunnel & Terminal R. R., 32d St. and Seventh Ave., New York, N. Y., and 76 Watson Ave., East Orange, N. J.

HUNTER, John A. (1909), Steam Engr., Am. Sheet & Tin Plate Co., Frick Bldg., Pittsburg, and Dickson Ave., Ben Avon, Pa.

HUTTON, Mancius S. (Junior, 1908), Junior Salesman, Am. Radiator Co., Bundy Dept., 129 Federal St., and *for mail*, 172 Huntington Ave., Boston, Mass.

HVID, Rasmus M. (1907; Associate, 1909), 29 Market St., Bethlehem, Pa.

KEITH, Robert R. (Junior, 1904), Asst. Genl. Supt., Light Fuel Oil Pump Co., and *for mail*, 687 Farwell Ave., Milwaukee, Wis.

KNIGHT, Hervey S. (Associate, 1898), Pat. Lawyer and Expt., 726 Ninth St., N. W., and 30 Piney Branch Rd., Washington, D. C.

LANE, Henry Marquette (1900), Editor, Castings and Wood Craft, Caxton Bldg., and *for mail*, 10613 Greenlawn Ave., Cleveland, O.

LARKIN, A. C. (1895; 1905), Babcock & Wilcox, Ltd., College St., St. Henry, Montreal, Canada.

LEWIS, John Ernest (Junior, 1909), Paonia, Colo.

LILLIBRIDGE, Ray D. (Associate, 1907), 195 Broadway, and P. O. Box 824, New York, N. Y.

McARTHUR, Arthur Royal (1906), Resident Engr., Am. Sheet & Tin Plate Co., and *for mail*, 674 Harrison St., Gary, Ind.

MARBURG, Louis Chas. (1909), 1777 Broadway, New York, N. Y.

MORLEY, Ralph (Junior, 1906), Mech. Engr., Transmission Dept., The Fairbanks Co., New York, N. Y., and *for mail*, 153 Delavan Ave., Newark, N. J.

NEWCOMB, Chas. L., Jr. (Associate, 1908), Denver Rock Drill & Mchy. Co., 18th and Blake Sts., Denver, Colo.

PARSONS, W. Everett (1899), Cons. Engr., 12 Bridge St., New York, and *for mail*, 10 Rich Ave., Mt. Vernon, N. Y.

PERRY, Samuel B. (Junior, 1895), Ins. Engr., 68 William St., New York, and Hollis, L. I., N.Y.

PHELPS, Charles C. (Junior, 1909), Editor, Steam, 2108 West St. Bldg., New York, N. Y.

POSEY, James (Junior, 1907), Cons. Engr., Painter & Posey, Cons. Engrs., 324 N. Charles St., Baltimore, Md.

RAY, Frederick (Junior, 1903), Ch. Engr., Alberger Pump Co., 140 Cedar St., New York, N. Y., and 19 Wilcox Place, East Orange, N. J.

REEDER, Nathaniel S., Jr. (1902; 1907), West Steel Car & Fdy. Co., 1470 Old Colony Bldg., Chicago, Ill.

ROGERS, Robert W. (Junior, 1908), Erie R. R., and *for mail*, 216 Walnut St., Meadville, Pa.

SALTZMAN, Auguste L. (1908), M.E., Asst. Ch. Engr., Edison Cos., Edison Laboratory, Orange, and *for mail*, 41 Watson Ave., East Orange, N. J.

SAMPSON, Chas.C. (1909), M. E., Supt. Constr., 5-8 Blowing Eng., Illinois Steel Co., South Chicago, and *for mail*, 7318 Champlain Ave., Chicago, Ill.
SANDO, Will J. (1899), Manager, 1908-1911; 430 Kane Pl., Milwaukee, Wis.
SCOTT, Walter G. (Junior, 1909), Cyclone Drill Co., Orville, O.
SEYMOUR, Dudley S. (1905), Supt., Union Spec. Mch. Co., 300 W. Kinzie St., Chicago, and *for mail*, 228 N. Elmwood Ave., Oak Park, Ill.
SMITH, Harry Ernest (1897), Chem. and Engr. of Tests, L. S. & M. S. Ry., Collinwood, and *for mail*, 36 Beersford Place, East Cleveland, O.
SMITH, J. Waldo (1896), Ch. Engr., Board of Water Supply, City of N. Y., 165 Broadway, and 136 Madison Ave., New York, N. Y.
STEENSTRUP, Peter Severin (1906), Box 1843, Seattle, Wash.
SYMONDS, George P. (1908), Chief Engrg. Dept., Alberger Condenser Co., 140 Cedar St., New York, N. Y.
TADDIKEN, J. F., Jr. (Junior, 1907), Am. Beet Sugar Co., Chino, Cal.
TERWILLIGER, Harry L. (Associate, 1901), Sales Mgr., Harron, Rickard & McCone, 139-149 Townsend St., San Francisco, and *for mail*, 1121 Emerson St., Palo Alto, Cal.
ULRICH, Max Julius (1906), Ch. Draftsman, Alberger Condenser Co., 140 Cedar St., New York, N. Y.
WADSWORTH, Frank L. O. (1903), Cons. Engr., 1347-1348 Oliver Bldg., and Duquesne Club, Pittsburg, Pa.
WESTERFIELD, George Sumner (Junior, 1903), Mgr. and Dist. Mgr., Hooven, Owens, Rentschler Co., Warren Webster & Co., B. F. Sturtevant Co., 326-329 Hennen Bldg., and 1320 Eleonore St., New Orleans, La.
WHITE, James A. (Junior, 1900), Genl. Elec. Co., and *for mail*, 19 Red Rock St., Lynn, Mass.
WHITEFORD, James F. (1908), Santa Fe Shops, Topeka, Kan.
WILLIAMSON, Leroy A. (Associate, 1902), Board of Trade Bldg., 131 State St., Boston, Mass.
WILSON, Wm. R. (Junior, 1899), Alberger Condenser Co., 140 Cedar St., New York, and *for mail*, 224 Palisade Ave., Yonkers, N. Y.
WINSHIP, James G. (1891), Internatl. Steam Pump Co., 115 Broadway, New York, and *for mail*, 209 Ocean Ave., Brooklyn, N. Y.

NEW MEMBERS

HODGE, Wm. W. (Junior, 1909), Field Engr., Dodge & Day, Lewiston, Pa.

DEATHS

BARY, Mark, December 1909.
BLOOMBERG, Jonas H.
EMERSON, Ralph Waldo, April 13, 1910.
PARSONS, William N., April 24, 1910.
PLUMMER, Frank J., April 15, 1910.
SPARROW, Ernest P., April 18, 1910.

GAS POWER SECTION

CHANGES OF ADDRESS

BIBBINS, James Rowland (1908), Mem.Am.Soc.M.E.

BIGELOW, Lucius S. (Affiliate, 1910), Pres., Light Pub. Co.,Pres.,Periodical's Pub. Co., 125 S. Main St., Willimantic, Conn.

HILLEBRAND, Herman (Affiliate, 1909), 638 W. Broad St., Bethlehem, Pa.

HOPCROFT, Ernest Bigly (Affiliate, 1908), L. W. Hall & Co., 50 Congress St., Boston, Mass.

MORLEY, Ralph (1908), Mem. Am.Soc.M.E.

RALSTON, Louis C. (Affiliate, 1909), R. F. D. 21, Box 41 A, San Jose, Cal.

SAMPSON, Chas. C. (1909), Mem.Am.Soc.M.E.

NEW MEMBERS

RIEPPEL, Paul (Affiliate, 1910), Blohm & Voss, Hamburg, Germany.

DEATHS

SPARROW, Ernest P., April 18, 1910.

STUDENT BRANCHES

CHANGES OF ADDRESS

GOLDSMITH, W. M. (Student, 1909), Greenwood Court, Greenwood Ave., Avondale, Cincinnati, O.
HESS, Harry L. (Student, 1909), Marysville, Cal.
LEVY, M. S. (Student, 1909), Metropole Hotel, Chicago, Ill.
MUDD, John P. (Student, 1909), 229 Zeralda St., Philadelphia, Pa.
SHULTS, L. J. (Student, 1909), 1820 S. Sawyer Ave., Chicago, Ill.
THOMAS, W. E. (Student, 1909), 4028 Sheridan Rd., Chicago, Ill.
WATSON, R. D. (Student, 1910), 237 Langdon St., Madison, Wis.

NEW MEMBERS

STEVENS INSTITUTE OF TECHNOLOGY

POLHEMUS, D. A. (Student, 1910), Stevens Inst. of Tech., Hoboken, N. J.

UNIVERSITY OF MAINE

BLAISDELL, A. H. (Student, 1910), 57 Fifth Ave., Bangor, Me.
COLE, R. F. (Student, 1910), Phi Beta Kappa House, Orono, Me.
CUMMINGS, C. G. (Student, 1910), Delta Tau Delta House, Orono, Me.
DANFORTH, H. N. (Student, 1910), Alpha Tau Omega House, Orono, Me.
HAMMOND, A. C. (Student, 1910), Main St., Orono, Me.
HARDY, S. J. (Student, 1910), Delta Tau Delta House, Orono, Me.
JOHNSON, C. A. (Student, 1910), Orono, Me.
LITTLEFIELD, P. H. (Student, 1910), Orono, Me.
MERRIAM, F. E. (Student, 1910), Orono, Me.
SCALES, E. M. (Student, 1910), Theta Epsilon House, Orono, Me.
SIMONTON, P. D. (Student, 1910), Orono, Me.

UNIVERSITY OF MISSOURI

EDGAR, O. N. (Student, 1910), 605 S. Fourth St., Columbia, Mo.
KENNEDY, F. T. (Student, 1910), Benton Hall, Columbia, Mo.
OLSEN, C. A. (Student, 1910), 411 S. Fifth St., Columbia, Mo.
PHILLIPS, E. C. (Student, 1910), 505 Conley Ave., Columbia, Mo.
PRICE, H. W. (Student, 1910), Lowry Hall, Columbia, Mo
SEXTON, C. E. (Student, 1910), 605 S. Fourth St., Columbia, Mo.
SHARP, H. N. (Student, 1910), 311 Waugh St.,Columbia, Mo.
STEED, A. (Student, 1910), Benton Hall, Columbia, Mo.
THACHER, F. B. (Student, 1910), Y. M. C.A. Bldg., Columbia, Mo.
WEAVER, H. E. (Student, 1910), 803 Virginia Ave., Columbia, Mo.
WESTCOTT, A. L. (Student, 1910), 1402 Windsor St., Columbia, Mo.

UNIVERSITY OF WISCONSIN

FALK, G. S. (Student, 1910), 627 Lake St., Madison, Wis.

COMING MEETINGS

May–June

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN EXPOSITION IN BERLIN

June 1–Aug. 31. American Manager, Max Vieweger, 50 Church St., New York.

AMERICAN BRASS FOUNDERS' ASSOCIATION

June 6–10, Detroit, Mich. Papers: Costs and Cost Systems, C. R. Stevenson; Analysis for Lead in Brass Alloys, C. P. Karr; Coöperative Course in Metallurgy, J. J. Porter; Electric Furnaces for Melting Non-Ferrous Alloys, A. L. Marsh; Fluxes as Applied to the Brass Foundry, I. S. Sperry; Electric Power as Applied to Melting, J. W. Richards, Mem. Am. Soc. M. E. Secy., W. M. Corse, Lumen Bearing Co., Buffalo, N. Y.

AMERICAN FOUNDRYMEN'S ASSOCIATION

June 6–10, Detroit, Mich. Secy. of general committee, A. Preston Henry, Standard Pattern Works.

ASSOCIATION OF CAR-LIGHTING ENGINEERS

June 7–8, semi-annual convention, Buffalo, N. Y. Secy., Geo. B. Colegrave, care of Central Railway, Chicago.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

June 22–24, summer meeting, Niagara Falls, N. Y. Secy., J. C. Olsen, Polytechnic Inst., Brooklyn.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

June 27–28, Annual Convention, Waumbek Hotel, Jefferson, N. H. Secy., R. W. Pope, 33 W. 39th St.

AMERICAN PORTLAND CEMENT MANUFACTURERS

June, Kansas City, Kans. Secy., P. H. Wilson, Land Title Bldg., Philadelphia, Pa.

AMERICAN RAILWAY ACCOUNTING OFFICERS

June 29, Colorado Springs, Colo. Secy., C. G. Phillips, 143 Dearborn St., Chicago.

AMERICAN RAILWAY MASTER MECHANICS ASSOCIATION

June 20–22, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

AMERICAN SOCIETY OF CIVIL ENGINEERS

June 1, 220 W. 57th St., New York. Papers: The New York Tunnel Extension of the Pennsylvania Railroad, B. H. M. Hewett; The North River Tunnels, W. L. Brown. June 21–24, Annual Convention, Chicago, Ill. Secy., C. W. Hunt, New York.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

May 27, St. Louis, with coöperation of Engineers Club of St. Louis. May 31–June 3, Spring Meeting, Atlantic City, N. J. July 26–29, meeting with Institution of Mechanical Engineers, in Birmingham and London, England. Secy., Calvin W. Rice, 29 W. 39th St., New York.

AMERICAN SOCIETY FOR TESTING MATERIALS

June 28-July 2, annual meeting, Atlantic City, N. J. Secy., Edgar Marburg, University of Pennsylvania, Philadelphia.

CANADIAN ELECTRICAL ASSOCIATION

July 6–8, annual convention, Royal Muskoka, Lake Rosseau. Secy., T. S. Young, Confederation Life Bldg., Toronto, Ont.

CANADIAN GAS ASSOCIATION

June 9–11, annual convention, Alexandra Rink, Hamilton, Ont. Secy., A. W. Moore, Woodstock, Ont.

CLEVELAND ENGINEERING SOCIETY

June 14, annual meeting, 714 Caxton Bldg. Secy., J. C. Beardsley.

ENGINEERS' CLUB OF BALTIMORE

June 4, annual meeting. Secy., R. K. Compton, City Hall.

ENGINEERS SOCIETY OF MILWAUKEE

June 8, annual meeting, Builders Club. Secy., W. F. Martin, 456 Broadway.

ENGINEERS SOCIETY OF PENNSYLVANIA

June 7, annual meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher, P. O. Box 704.

FREIGHT CLAIM ASSOCIATION

June 15, Los Angeles, Cal. Secy., W. P. Taylor, Richmond, Va.

INTERNATIONAL CONGRESS OF INVENTORS

June 13–18, Rochester, N. Y.

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

Last week in June, Düsseldorf, Prussia. Secy., Dr. E. Schrödter, Jacobstrasse 315.

IOWA DISTRICT GAS ASSOCIATION

June 15–17, annual meeting, Sioux City. Secy., G. I. Vincent, Des Moines.

MANUFACTURERS' SUPPLY ASSOCIATION

June 6–10, exhibit, Detroit, Mich. Secy., C. E. Hoyt, Lewis Institute, Chicago.

MASTER CAR BUILDERS ASSOCIATION

June 15–17, Atlantic City, N. J. Secy., J. W. Taylor, 390 Old Colony Bldg., Chicago.

NATIONAL DISTRICT HEATING ASSOCIATION

June 1–3, annual meeting, Toledo, O. Secy., D. C. Gaskill, Greenville, O.

NATIONAL ELECTRIC CONTRACTORS' ASSOCIATION

July 20, annual meeting, Atlantic City, N. J. Secy., W. H. Morton, Martin Bldg., Utica, N. Y.

NATIONAL ELECTRIC TRADES ASSOCIATION

June, San Francisco, Cal. Secy., F. B. Vose, 1343 Marquette Bldg., Chicago.

NATIONAL GAS AND GASOLINE ENGINE TRADES ASSOCIATION
June 13–16, semi-annual meeting, Hotel Sinton, Cincinnati, O. Subjects for discussion: Carbureters, Geo. M. Schebler; Ignition in Gas and Gasoline Engines, Carl Pfanstiehl; Gas Producers, L. F. Burger; Present and Future Opportunities in the South for the Gas Engine Boiler, W. R. C. Smith; How to Illustrate and Describe Mechanical Installations, O. Monnett; Heavy Hitting in the Advertisers' League, Ren Mulford, Jr.; Some Association Experiences, F. J. Alvin; Dry Batteries, H. S. Green; The Gas Engine Field in Mexico, G. W. Hall; Large Gas Engines, J. D. Lyon. Secy., Albert Stritmatter.

NATIONAL SOCIETY FOR PROMOTION ENGINEERING EDUCATION
June 23-25, annual meeting, Madison, Wis. Papers on Technical Education Abroad; Inspection Trips for Technical Students, Efficiency in Technical Education. Secy., Prof. H. H. Norris, Cornell University, Ithaca, N. Y.

NEW ENGLAND WATERWORKS ASSOCIATION
June, Providence, R. I. September 14–16, annual convention, Rochester N. Y. Secy., Willard Kent, Narragansett Pier, R. I.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS
June 28, annual meeting. Secy., T. M. Phetteplace, Mem.Am.Soc.M.E., 48 Snow St.

RAILWAY SIGNAL ASSOCIATION
June 14, 29 W. 39th St., New York, 9.30 a.m. Secy., C. C. Rosenberg, Bethlehem, Pa.

RENSSELAER SOCIETY OF ENGINEERS
June, annual meeting, Rensselaer Polytechnic Inst., 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

TELEPHONE SOCIETY OF NEW YORK
June 21, annual meeting, 29 W. 39th St. Secy., T. H. Woolhouse.

TRANSPORTATION AND CAR ACCOUNTING OFFICERS
June 28. Secy., G. P. Conard, 24 Park Pl., New York.

MEETINGS IN THE ENGINEERING SOCIETIES BUILDING

Date	Society	Secretary	Time
June			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
3	Western Union Electrical Society	H. C. Northen	7.00
4	Amer. Soc. Hun. Engrs and Archts	E. L. Mandel	8.30
9	Illuminating Engineering Society	P. S. Millar	8.15
10	Western Union Electrical Society	H. C. Northen	7.00
			a.m.
14	Railway Signal Association	C. C. Rosenberg	9.30
			p.m.
17	Western Union Electrical Society	H. C. Northen	7.00
21	New York Telephone Society	T. H. Lawrence	8.00
24	Western Union Electrical Society	H. C. Northen	7.00
July			
7	Blue Room Engineering Society	W. D. Sprague	8.00

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Geo. M. Bond ....Hartford, Conn.
R. C. Carpenter ....Ithaca, N. Y.
F. M. Whyte ....New York

Terms expire at Annual Meeting of 1910

Charles Whiting Baker ....New York
W. F. M. Goss ....Urbana, Ill.
E. D. Meier ....New York

Terms expire at Annual Meeting of 1911

PAST PRESIDENTS

Members of the Council for 1910

John R. Freeman ....Providence, R. I.
Frederick W. Taylor ....Philadelphia, Pa.
F. R. Hutton ....New York
M. L. Holman ....St. Louis, Mo.
Jesse M. Smith ....New York

MANAGERS

Wm. L. Abbott ....Chicago, Ill.
Alex. C. Humphreys ....New York
Henry G. Stott ....New York

Terms expire at Annual Meeting of 1910

H. L. Gantt ....Pawtucket, R. I.
I. E. Moultrop ....Boston, Mass.
W. J. Sando ....Milwaukee, Wis.

Terms expire at Annual Meeting of 1911

J. Sellers Bancroft ....Philadelphia, Pa.
James Hartness ....Springfield, Vt.
H. G. Reist ....Schenectady, N. Y.

Terms expire at Annual Meeting of 1912

TREASURER

William H. Wiley ....New York

CHAIRMAN OF THE FINANCE COMMITTEE

Arthur M. Waitt ....New York

HONORARY SECRETARY

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SECRETARY

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H. L. Gantt
F. M. Whyte

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FINANCE

Arthur M. Waitt (5), *Chairman*
Robert M. Dixon (3), *Vice-Chairman*
Edward F. Schnuck (1)
Geo. J. Roberts (2)
Waldo H. Marshall (4)

HOUSE

William Carter Dickerman (1) *Chairman*
Francis Blossom (3)
Bernard V. Swenson (2)
Edward Van Winkle (4)
H. R. Cobleigh (5)

LIBRARY

John W. Lieb, Jr. (3), *Chairman*
Leonard Waldo (2)
Ambrose Swasey (1)
Chas. L. Clarke (4)
Alfred Noble (5)

MEETINGS

Willis E. Hall (5), *Chairman*
L. R. Pomeroy (2)
Wm. H. Bryan (1)
Chas. E. Lucke (3)
H. de B. Parsons (4)

MEMBERSHIP

Charles R. Richards (1) *Chairman*
George J. Foran (3)
Francis H. Stillman (2)
Hosea Webster (4)
Theo. Stebbins (5)

PUBLICATION

D. S. Jacobus (1) *Chairman*
Fred R. Low (3)
H. F. J. Porter (2)
Geo. I. Rockwood (4)
Geo. M. Basford (5)

RESEARCH

W. F. M. Goss (4), *Chairman*
R. H. Rice (2)
R. C. Carpenter (1)
Ralph D. Mershon (3)
Jas. Christie (5)

Note—Numbers in parentheses indicate number of years the member is yet to serve.

SPECIAL COMMITTEES

1910

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
A. P. Trautwein — Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet — H. H Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman* — F. R. Hutton
G. M. Basford — D. S. Jacobus
Jesse M. Smith

On Conservation of Natural Resources

Geo. F. Swain, *Chairman* — L. D. Burlingame
Charles Whiting Baker — M. L. Holman
Calvin W. Rice

On International Standard for Pipe Threads

E. M. Herr, *Chairman* — Geo. M. Bond
William J. Baldwin — Stanley G. Flagg, Jr.

On Standards for Involute Gears

Wilfred Lewis, *Chairman* — E. R. Fellows
Hugo Bilgram — C. R. Gabriel
Gaetano Lanza

On Power Tests

D. S. Jacobus, *Chairman* — L. P. Breckenridge — Edward F. Miller
Edward T. Adams — William Kent — Arthur West
George H. Barrus — Charles E. Lucke — Albert C. Wood

On Student Branches

F. R. Hutton, Honorary Secretary

On Meetings of the Society in Boston

Ira N. Hollis, *Chairman* — I. E. Moultrop, *Secretary*
Edward F. Miller — J. H. Libbey
Charles T. Main

On Meetings of the Society in St. Louis

Wm. H. Bryan, *Chairman* — Earnest L. Ohle, *Secretary*
R. H. Tait, *Vice-Chairman* — M. L. Holman
Fred E. Bausch

On Arrangements for Joint Meeting in England

Ambrose Swasey, *Chairman* — Chas. Whiting Baker, *Vice-Chairman*
Geo. M. Brill — F. R. Hutton
John R. Freeman — Willis E. Hall
W. F. M. Goss — Calvin W. Rice
George Westinghouse — Wm. H. Wiley

SOCIETY REPRESENTATIVES

1910

On John Fritz Medal

AMBROSE SWASEY (1) — CHAS. WALLACE HUNT (3)
F. R. HUTTON (2) — HENRY R. TOWNE (4)

On Board of Trustees United Engineering Societies Building

F. R. HUTTON (1) — FRED J. MILLER (2)
JESSE M. SMITH (3)

On Library Conference Committee

J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE, AM. SOC. M. E.

On National Fire Protection Association

JOHN R. FREEMAN — IRA H. WOOLSON

On Joint Committee on Engineering Education

ALEX. C. HUMPHREYS — F. W. TAYLOR

On Advisory Board National Conservation Commission

GEO. F. SWAIN — JOHN R. FREEMAN
CHAS. T. MAIN

On Council of American Association for the Advancement of Science

ALEX. C. HUMPHREYS — FRED J. MILLER

NOTE—Numbers in parentheses indicate number of years the member is yet to serve.

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1909-1910

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J. R. Bibbins	Geo. A. Orrok

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G. I. Rockwood (2)
H. H. Suplee (4)
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GAS POWER MEMBERSHIP COMMITTEE

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H. V. O. Coes
G. M. S. Tait
A. E. Johnson
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Nisbet Latta
E. D. Dreyfus
C. W. Obert
C. T. Wilkinson

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L. N. Ludy
A. L. Rice
A. J. Wood

GAS POWER INSTALLATIONS COMMITTEE

L. B. Lent, *Chairman*
A. Bement
C. B. Rearick

GAS POWER PLANT OPERATIONS COMMITTEE

I. E. Moultrop, *Chairman*
C. N. Duffy
J. D. Andrew
H. J. K. Freyn
C. J. Davidson
W. S. Twining
C. W. Whiting

GAS POWER STANDARDIZATION COMMITTEE

C. E. Lucke, *Chairman*
E. T. Adams
Arthur West
James D. Andrew
J. R. Bibbins
H. F. Smith
Louis C. Doelling

Note—Numbers in parentheses indicate number of years the member is yet to serve.

OFFICERS OF STUDENT BRANCHES

INSTITUTION	BRANCH AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
	1908			
Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	C. C. Allen	C. F. Hirschfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	G. F. Gebhardt	F. E. Wernick	W. E. Thomas
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	A. F. Meston	J. B. Bubb
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College. Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. W. Templin	H. A. Houston
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	C. E. Johnson	C. A. Swiggett
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	B. L. Keown	C. S. Huntington
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	Chas. E. Lucke	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Morril Mackenzie	Foster Russell
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	W. H. Montgomery	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	John S. Langwell	Karl L. Kraatz
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. V. Aycock	Osmer Edgar
Univ. of Nebraska. Lincoln, Neb.	December 7	C. R. Richards	M. E. Strieter	A. D. Stancliff
	1910			
Univ. of Maine, Orono, Me.	February 8	Arthur C. Jewett....	H. N. Danforth	A. H. Blaisdell
Univ. of Arkansas, Fayetteville, Ark.	April 12	B. N. Wilson	C. B. Boles	W. Q. Williams

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CONTENTS

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THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 JANUARY 1910 NUMBER 1

THE New York monthly meeting of the Society will be held in the Engineering Societies' Building on Tuesday evening, January 11. The subject for discussion is Lubrication. The paper upon Efficiency Tests of Lubricating Oils by Prof. F. H. Sibley of the University of Alabama, published in The Journal for November, will be presented and important contributions upon the properties of lubricants, their efficiency, durability, characteristics, etc., will be made by Dr. C. F. Mabery, of Case School, Cleveland, and Genl. Chas. Miller of Franklin, Pa.

Dr. Mabery has been engaged for a long period of time in experiments upon lubricating oils and has obtained results of unusual interest, because of the uniformity attained in repeating experiments, always a difficult matter in testing lubricants. General Miller has been so long identified with the subject of lubrication and has so large a fund of information as a result of this experience that his remarks will add greatly to the interest of the evening. There will be discussions also by F. R. Low, Editor of Power, I. E. Moultrop, mechanical engineer of the Boston Edison Company, J. P. Sparrow, chief engineer of the New York Edison Company, and others.

The subject of lubrication is so important in its bearing upon the conservation of power and upon machinery of all kinds, especially since the introduction of recent new types, such as the steam turbine and automobile, that it is desirable to have authentic information easily available for the use of engineers. By introducing the subject for discussion before the Society, it is hoped that this result may eventually be brought about and that a substantial beginning will be made at this meeting.

MEETING IN ST. LOUIS, JANUARY 15

The next monthly meeting of the Society in St. Louis will be held on January 15. The usual announcement of this meeting with details in regard to the paper and discussion will be sent to members and engineers in St. Louis and vicinity previous to the meeting.

MEETING IN BOSTON, JANUARY 21

A joint meeting of The American Society of Mechanical Engineers, the Boston Society of Civil Engineers and the Boston branch of the American Institute of Electrical Engineers, will be held in Boston on the evening of January 21. Committees have been appointed by the society of civil engineers and the local section of the electrical engineers to coöperate with the local committee of this Society to complete arrangements. The meeting will take the form of a banquet and reception, with the presidents of the three societies in attendance, George H. Westinghouse of The American Society of Mechanical Engineers, L. B. Stilwell of the American Institute of Electrical Engineers and George B. Francis of the Boston Society of Civil Engineers, besides the incoming president of the American Society of Civil Engineers, John A. Bensel, and other distinguished guests. The banquet hall of the Hotel Somerset, which is the largest and finest in the city has been engaged for the occasion.

Following the banquet there will be addresses by some of the guests and a paper on the Main and Auxiliary Machinery of the Battleship North Dakota, illustrated with lantern slides, by Charles B. Edwards of the Fore River Shipbuilding Company. There is under discussion at Boston a project for building and equipping a united engineering building and the president of the Boston Society of Civil Engineers will bring up this subject and describe what efforts that Society has already made towards this end.

The meetings of The American Society of Mechanical Engineers in Boston have been uniformly well attended, as have those of the other societies, and it is believed that this joint meeting will bring together an unusually large number of engineers and that it will be the most successful similar meeting of the kind that has taken place in that city.

SPRING MEETING, ATLANTIC CITY, MAY 31–JUNE 6

The Spring Meeting of The American Society of Mechanical Engineers will be held this year as usual, in addition to the London Meet-

ing which occurs in July. Atlantic City has been selected by the Meetings Committee and approved by the Council as the place and the time will be from May 31–June 6, inclusive. The headquarters during the meeting will be at the Marlborough-Blenheim Hotel.

JOINT MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND THE INSTITUTION OF MECHANICAL ENGINEERS

In response to the invitation of The Institution of Mechanical Engineers of Great Britain, received and accepted by The American Society of Mechanical Engineers, and recently sent out to the general membership, 133 members and 100 ladies have signified their intention of attending the joint meeting in Great Britain in the summer of 1910, and 183 have expressed themselves as giving the matter favorable consideration.

The present indications are that some of the functions will be held in Manchester, Birmingham or Sheffield, possibly concluding in London, and the invitation itself is an earnest of the notable professional and social opportunities which will be extended to the Society. Arrangements will probably be made for the accommodation of the members on the same steamer.

Where time and personal engagements permit, the visiting members will have the opportunity of attending the following events and meetings which are to take place during the summer of 1910: Anglo-Japanese Exhibition at Sheperds Bush, London; American Exposition in Berlin; Brussels Universal and International Exhibition; London Pageant, probably at Chester; Pageant at Bristol; Church Pageant at Fulham Palace; Military Pageant in London; International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, at Düsseldorf; International Sports Exhibition at Vienna; International Exhibition of Arts and Industries, Alexandra Palace, London; the Passion Play at Oberammergau.

It is expected that papers will be presented by members of both societies on electrification of railways, on round house practice and the handling of locomotives at terminals, on certain phases of machine shop practice, and on the subject of standards for gear teeth which is now being considered by a committee of this Society as well as by a committee of the Institution of Mechanical Engineers. While papers will be mainly restricted to the subjects indicated, the Meetings Committee will be pleased to consider papers on other subjects.

REPORTS OF MONTHLY MEETINGS

BOSTON MEETING, NOVEMBER 17

A very successful meeting of the Society was held at Boston in the Lowell Building, Massachusetts Institute of Technology, Wednesday evening, November 17. Two hundred and forty were present at this meeting and the Low-pressure Steam Turbine was the topic of discussion.

Mr. Henry G. Stott of the Interborough Rapid Transit Company gave an interesting account of the difficulties encountered as well as the very fine results obtained from an installation recently made at the 59th Street Station of his company, New York. Mr. W. L. R. Emmet described the low-pressure turbine situation from his viewpoint and pointed out the advantages of this type of prime mover for many mill installations and industrial works in New England. Mr. H. E. Longwell, consulting engineer of the Westinghouse Machine Company, and Edward L. Clark, manager of their Boston office, both spoke on the work that company are doing in this field. Mr. Max Rotter, turbine engineer of the Allis-Chalmers Company, pointed out in a humorous way a number of situations where the low-pressure turbine was not a desirable proposition. Professor Miller of the Massachusetts Institute of Technology also discussed the subject.

BOSTON MEETING, DECEMBER 17

On Friday evening, December 17, a goodly number of engineers of Boston and vicinity gathered at the call of the local members of The American Society of Mechanical Engineers to discuss the Effect of Superheated Steam on Cast Iron. The meeting was called to order by Prof. Ira N. Hollis, who announced that the next meeting would be held on January 21 and would take the form of a reception, possibly a complimentary dinner, to the newly elected president of the American Society, George H. Westinghouse, and other of the Society's officials. A later announcement of this meeting is contained elsewhere in this number, and of the coöperation of the Boston Society

of Civil Engineers and of the local members of the Institute of Electrical Engineers. The committee which has been in charge of the meetings, consisting of Messrs. Hollis, Moultrop, Miller, Mann and Libbey, was continued.

The set papers which were published in the December issue of The Journal were then presented by their authors—Prof. Edward F. Miller of Boston, Arthur S. Mann of Schenectady, and Prof. Ira N. Hollis of Boston in the order named, and were discussed by Messrs. Collins of Stone & Webster, George A. Orrok of the New York Edison Company, Chas. H. Bigelow of Chas. T. Main's office, W. K. Mitchell of Philadelphia, Messrs. Primrose and Nutting of the Power Specialty Company, Wm. E. Snyder of the American Steel and Wire Company and others. The general purport of the discussion, was rather reassuring to the users of cast iron pipe and fittings, and to those who are interested in the extension of the use of superheated steam, in indicating that superheated steam *per se* has no injurious effect upon cast iron fittings, but that if the pipe lines are properly designed for the greater ranges of temperature, if the fittings are made adequate to the pressure and if fluctuations in temperature can be avoided, the use of superheated steam introduces no piping difficulties which can not be easily overcome.

MEETING AT ST. LOUIS, NOVEMBER 13

At the meeting of the Society at St. Louis, November 13, with the Engineers' Club at St. Louis, a description of the new plant of the Heine Safety Boiler Company of Boston was presented by E. R. Fish, under the title, A Modern Boiler Shop. There was also further discussion of Professor Carpenter's paper on High-Pressure Fire Service, continued from the October meeting.

MEETING AT ST. LOUIS, DECEMBER 11

A meeting was held with the Engineers' Club of St. Louis on Saturday evening, December 11, at the rooms of the latter society. The meeting was called to order by William H. Bryan, member of the Meetings Committee of the Society and chairman of the joint committee of the two societies at St. Louis. Prof. E. L. Ohle acted as secretary. There were present fifty-five members and guests.

The paper of the evening was by G. R. Parker of the General Electric Company, on The Relation of the Steam Turbine to Modern

Central Station Practice, in which the underlying principles of modern steam turbines were discussed, together with the design of various prominent types on the market, and the developments made in recent years in improving capacity and efficiency. Attention was called to the large turbine capacity which may now be obtained within limited floor space; to the question of low-pressure turbines and their availability in supplementing standard reciprocating engines, increasing both their capacity and economy; also to the work already done in this direction at the plant of the Union Electric Light & Power Company in St. Louis, and to prospective work along similar lines in the same plant. The address was illustrated by lantern slides.

Discussions followed by Chairman Bryan, Prof. H. W. Hibbard, L. R. Day, E. R. Smith and Prof. E. L. Ohle, in which many additional interesting points were brought out.

On the afternoon of the day of the meeting an excursion was made to the Ashley Street plant of the Union Electric Light & Power Company, for the inspection of the apparatus and equipment, on the invitation of John Hunter, chief engineer. This excursion, supplementing as it did the paper of the evening, added much to the interest and value of the meeting and a vote of thanks was extended to Mr. Hunter for the opportunity so generously afforded.

THE ANNUAL MEETING

The thirtieth annual meeting of The American Society of Mechanical Engineers was held in the Engineering Societies Building December 7 to 10, with an attendance of 628 members and 435 guests. This year, for the first time, the arrangements of the entertainment features were entirely in the hands of the local committee, the members in New York and vicinity acting as hosts, and the results fully justified this method of handling an important part of the annual meeting.

Despite the severe storm on Tuesday evening, the President's reception was well attended, and a large audience gathered in the auditorium. On Wednesday afternoon the trip through the Pennsylvania Terminal brought out a large body of members and guests, and in the evening L. W. Ellis, of the Bureau of Plant Industry, U. S. Dept. of Agriculture, delivered an interesting lecture on the Era of Farm Machinery. On Thursday evening the attendance at the reception in the magnificent ball room of the Hotel Astor was nearly 600.

OPENING SESSION, TUESDAY EVENING

The President's reception on Tuesday evening was undoubtedly one of the most enjoyable ever held, the members and guests comfortably filling the handsomely decorated rooms of the Society, though the inclement weather doubtless prevented a larger attendance.

The session was called to order in the auditorium by Vice-President Fred J. Miller, who presented President Jesse M. Smith. President Smith then proceeded with his address on The Profession of Engineering, which is printed in full in this number. It deals mainly with the need of coöperation among engineers, looking toward the maintenance of high standards in engineering practice.

Following the address, Theodore Stebbins, chairman of the Tellers of Election, presented to the President the report on the election of officers and the following were thereupon declared elected: For president, George Westinghouse; for vice-presidents, Charles Whiting

Baker, W. F. M. Goss, E. D. Meier; for managers, J. Sellers Bancroft, James Hartness, H. G. Reist; for treasurer, William H. Wiley.

President Smith then called on Past-Presidents Worcester R. Warner, Geo. W. Melville and Samuel T. Wellman to escort President-elect George Westinghouse to the platform.

After his notification of election and introduction to the members, the president-elect spoke as follows:

When Mr. Warner, the Chairman of your Nominating Committee, after first writing on the subject, came to Lenox to ask me to accept the nomination for president of this great Society, I had already decided that it would be impossible for me to have the privilege of accepting; but after he had explained to me the desires of his associates and had represented to me that it was the unanimous wish of all of the members of your Nominating Committee to honor me at this particular time, and in so doing to express an appreciation of my efforts and accomplishments in the engineering field, I with much hesitation consented to accept the nomination and promised if elected to do everything in my power.

Whether two mistakes have been made—one in yielding to the persuasive words of Mr. Warner, and the other in my election as your president—the forthcoming year will determine. I trust I may be able to fulfil your expectations by adding something to the worldwide reputation of The American Society of Mechanical Engineers.

With these remarks, I now accept with feelings of deep gratitude the honor which the members of the Society have tonight unanimously conferred upon me.

There never was a time in the history of the world when honest, wise and conservative action is more strongly demanded of us and of all men than now, if we have any desire to preserve the right to comfortably carry on our various affairs.

I thank you, and I ask your coöperation in my efforts to perform my duties as your president.

The meeting was then adjourned to the rooms of the Society where the members and guests were introduced by Secretary Calvin W. Rice, to the President-elect and Mrs. Westinghouse, those also in the receiving line being President Jesse M. Smith and Mrs. Smith, Mrs. Hutton and Honorary Secretary F. R. Hutton.

WEDNESDAY EVENING LECTURE

As already stated the lecture on Wednesday evening was on the Era of Farm Machinery, by L. W. Ellis, of the Bureau of Plant Industry of the United States Department of Agriculture at Washington, D. C. The lecture was illustrated by lantern slides. Mr. Ellis first gave an idea of agricultural progress, by describing some of the most striking mechanical achievements found on Western farms of the present day. He first described early farm implements and told briefly of the

transition from hand to machine methods. In 1800 wheat was sown broadcast by hand, after the ground had been plowed with a heavy, clumsy, wooden plow, requiring as many as eight oxen to pull it. Sickles cut the grain, and it was bound by hand. During the succeeding winter it was threshed out either by a flail or by driving animals over it as it lay in heaps. It was finally winnowed by hand.

Corn cultivation was by the hoe, or a rude shovel plow. The stalks were cut and the ears husked out by hand. Shelling was done by scraping the ears against the handle of a frying pan—a bushel in one hundred minutes.

Hay was cut with a scythe and was pitched by hand from ground to cart, and cart to haymow. Baling and shipping were practically unknown. Hand methods prevailed in the dairy, the stable, the cotton fields, the potato patch—in fact in every phase of production.

From 1855 to 1894 the human labor consumed in producing a bushel of corn by the best available methods declined from four hours and thirty minutes to forty-one minutes, and for shelling it from one hundred minutes to one minute. In 1830, three hours and three minutes of human labor were required to raise and thresh a bushel of wheat—in 1896 ten minutes. Eleven hours were required to cut and cure a ton of hay in 1860, and but one hour and thirty-nine minutes in 1894.

Power corn shellers now used have a capacity of from one hundred to eight hundred bushels per day. The cobs are carried to a pile and the shelled corn delivered into sacks or wagons. The fuel value of the cobs pays the cost of shelling.

Though hand methods still prevail in some sections, the mower is now practically the universal means of cutting the hay crop. This is a modification of the early reaping machines with such factors eliminated as are not necessary for cutting the grass. The steel self-dump rake, the side-delivery rake and the hay loader, the stacker, and the bailing press are other developments for hay harvesting.

In the extreme West there has been developed the combined harvester which seems to represent the greatest possible saving of human labor. This machine, drawn by from twenty to forty horses, under control of a single driver, cuts, threshes, recleans, and delivers into sacks the grain from forty to fifty acres per day. Two men are required for sewing the sacks. The straw, including all weed seeds, is distributed over the ground as the team proceeds. On level land the horses may be replaced by the steam engine, which furnishes power sufficient to cut a swath up to forty feet in width and to cover from seventy-five to one hundred and twenty-five acres per day.

For general farm work the internal-combustion tractor may be said to be rapidly supplanting the steam engine, which, however, has a great field of usefulness in sections where it is desired to bring large areas rapidly under cultivation. In older sections, in order to compete successfully with the horse, tractors must bring the cost of operation close to the cost with horses and at the same time be capable of a great variety of work. The internal-combustion tractor meets these conditions better than the steam engine, and is being introduced at a rate estimated anywhere from two thousand to five thousand per year.

The automobile is rapidly finding a place in the business management of the farm. It takes from the heavy draft horse the necessity for long, exhausting trips to town on light errands.

In general, machinery has reduced the cost of producing farm products. It has improved the quality of products by condensing crop operations within the period when the most favorable conditions prevail. By increasing the acre effectiveness of a man it has reduced the labor necessary to produce the nation's food supply, leaving it free to assist in development along other lines. At the same time it has thrown upon the cities the burden of providing work for an ever increasing army of non-producers. It has increased the investment necessary for the proper organization of a farm, this and the price of land making it more difficult for a person of small capital to engage in farming.

As a nation we have occupied nearly all of our naturally productive area and are confronted with the necessity of providing food for an increasing population with a constant acreage. In the past, machinery has encouraged extensive rather than intensive farming. Henceforth the reverse should be true. If he who makes two blades of grass grow where one grew before, is a public benefactor, then none the less is he a public servant who puts into the farmer's hands the machinery for making such a course attractive.

BUSINESS MEETING

The business session on Wednesday morning was called to order by President Jesse M. Smith. Secretary Calvin W. Rice read the annual report of the Council. The Secretary then read the report of the Tellers of Election of members, which will be published in the membership list of the Society. The list included 166 applicants for membership and 21 for advance in grade.

The next in order was the consideration of the proposed amendments to the Constitution. The first amendment relates to C 10 on associate membership, which reads as follows:

C 10 An Associate shall be 26 years of age or over. He must either have the other qualifications of a member or be so connected with engineering as to be competent to take charge of engineering work, or to coöperate with engineers.

The proposed amendment reads as follows:

An associate member shall be thirty years of age or over; he must have been so connected with some branch of engineering, or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge.

Another amendment relates to the clause on Junior Membership which now reads as follows:

C 11 A Junior shall be 21 years of age or over. He must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school.

The following addition is proposed by the Committee on Constitution and By-Laws:

A person who is over 30 years of age can not enter the Society as a Junior.

Both these amendments have been approved by the Committee on Membership. It therefore remains for the members to vote on them by letter ballot.

A third proposed amendment to the Constitution relates to the formation of an additional standing committee. This was presented at the Washington meeting in the form of a resolution, as follows:

Resolved, That we recommend to the Council the appointment of a Public Relations Committee, to investigate, consider and report on the methods whereby the Society may more directly coöperate with the public on engineering matters and on the general policy which should control such coöperation.

It was moved and seconded that this also be referred to the members for letter ballot.

Dr. D. S. Jacobus, Chairman of the Committee on Power Tests, then made a verbal report. This committee was appointed to revise all the codes relating to power tests, some of which did not agree with others, or were not up to date. It had been decided to blend the whole into one report rather than present a series of reports, as on engine testing, boiler testing, etc. The first part of the report will deal with tests in general, calibration of apparatus, units, etc., while

the second part will be subdivided for the various classes of machines and apparatus.

Geo. H. Barrus had volunteered to prepare a skeleton of the report and had done excellent work in this respect, the material making 69 closely type-written pages. Copies of this outline were in the hands of the members of the committee and would shortly be discussed by them.

Dr. Jacobus also made a verbal report for the Joint Committee on a Standard Tonnage Basis for Refrigeration. This committee had made a preliminary report in 1904 and suggested certain units for measuring the refrigerating capacity of the machinery. They had also suggested a standard set of conditions under which a machine should be tested to obtain the refrigerating capacity of that machine. Later on, the work of the committee was extended, and they were asked to recommend a method of testing the machines. A preliminary report was also prepared on this portion of the work and had been before the Society.

Though the committee had received some favorable discussion on the report they felt that it was not a complete piece of work, and they wished that some one would give the committee additional light on how the report could be made. Furthermore, there were many places in the report where the committee could not make any definite recommendations, because they did not have enough data at hand.

A résumé of the work that has been done by the Committee on Refrigeration was prepared and sent to the Congress of Refrigerating Industries, held in Paris in the fall of 1908, with the request that it be discussed. In making this résumé certain questions were asked, on which the committee wished to obtain specific information. This was done in a semi-official way, and after taking up the matter with the Secretary of this Society, the committee ended the communication to the International Committee in this way:

The policy of The American Society of Mechanical Engineers has always been for the advancement of the arts, and whereas it is only natural that it should take pride in participating in advancements, it will never look except with satisfaction upon activities of other bodies, even in the subjects on which it has worked.

I feel safe in saying, therefore, that any criticism by the members of this organization on the work which has been done in connection with the subject at hand will be gladly received. Criticism leads to the establishment of better and more up-to-date methods, and what The American Society of Mechanical Engineers is after, and what I am sure we are all after, is to work hand in hand for the good of the cause.

I also feel safe in saying that The American Society of Mechanical Engineers

will coöperate in every way in the endeavor to establish some standard set of rules which shall conform with the views of such able experts as are gathered in this meeting. It is certainly hoped that the matter presented in this paper will receive a thorough discussion, irrespective of whether those who take part agree or disagree with the findings of the committee.

About the same time, a request was made by the committee that it should be allowed to coöperate with a committee of the American Society of Refrigerating Engineers, so that if this general committee recommended certain units, they would really be used by both Societies. A committee of five was appointed by the American Society of Refrigerating Engineers to coöperate with the committee of five of The American Society of Mechanical Engineers. This combined committee has already held one meeting and sent out a circular letter to a number of refrigerating engineers, reviewing the units that had been recommended by the Society, and asking for an opinion regarding these specific units. A great number of replies had been received, showing how much interest there is in the subject. Most of the replies said either that the units were acceptable to those who had read the letter, or that they would leave the selection of the units entirely in the hands of the committee. The committee therefore has a very good working basis, and hopes within a comparatively short time to be able to present the results of its work.

Dr. C. E. Lucke then abstracted the report of the Gas Power Standardization Committee, of which he is chairman. The report was discussed by Dr. D. S. Jacobus, Prof. R. H. Fernald, A. A. Cary, Edwin D. Dreyfus and L. B. Lent.

The report of the Gas Power Plant Operations Committee was presented by F. R. Low in the absence of I. E. Moultrop, chairman of the committee. The report was discussed by Prof. R. H. Fernald, Edwin D. Dreyfus, and Arthur J. Wood.

THURSDAY MORNING SESSION

The Thursday morning session was devoted to papers on the measurement of the flow of fluids.

The first paper presented was on Tests on a Venturi Meter for Boiler Feed, by Prof. C. M. Allen, of Worcester Polytechnic Institute. The object of these tests with the venturi meter was to determine how well adapted it would be for use in measuring the feed to a boiler, in view of the variety of conditions under which it might have to operate. The methods of pumping the water through the meter, the different temperatures of the water pumped, various and fluctuating

pressures and velocities of flow, any one or several of these conditions might be met in actual service, and the results obtained indicate that such occurrence would have practically no effect on the satisfactory performance of the work of the meter. Though there are limits to the satisfactory operation of any one meter, the tests indicate that the venturi meter is sufficiently accurate for the majority of commercial or engineering requirements.

The paper was discussed by F. N. Connet and Clemens Herschel, Dr. Sanford A. Moss and Prof. L. S. Marks submitting written discussions.

The next paper, Efficiency Tests of Steam Nozzles, by Prof. F. H. Sibley, of the University of Alabama, was read by Prof. C. C. Thomas of the University of Wisconsin. The object of the test was to determine the efficiency of various shaped nozzles with steam flowing from a given initial pressure to a known vacuum; also to determine the effect on the efficiency of changing the angle of divergence. Two methods were tried out for finding this efficiency: (*a*) by first finding the pressure in the nozzle by means of a search tube placed axially in the nozzle; (*b*) by finding the reaction of the nozzle by suspending it in an air-tight box at the end of a flexible steel tube. The deflection of the tube caused by the reaction of the nozzle was measured by a calibrated spring. The results of the tests indicate: (*a*) that the reaction is affected by a difference in pressure between the muzzle of the nozzle and the medium surrounding the nozzle; (*b*) that the efficiencies of the various nozzles were determined within a probable error of 2 per cent; (*c*) that the efficiency is affected more by the smoothness of finish on the inside of the nozzle than by the exact contour of the nozzle.

A. F. Nagle, A. R. Dodge and Professor Thomas discussed the paper, J. A. Moyer submitting a written discussion.

George F. Gebhardt's paper on The Pitot Tube as a Steam Meter was read by the Secretary in the author's absence. The application of a pitot tube system as described in the paper is an accurate means of determining the *velocity* of steam at any point in a pipe, provided the values of the various influencing factors are known; and for straight lengths of piping with continuous flow, under these conditions, it is an accurate means of determining the *weight* of steam flowing. Under average commercial conditions in which the pressure and quality of the steam fluctuate and an average value must be taken for the density of the self-adjusting water column, only approximate results can be obtained, the extent varying with the degree of fluctuation.

Walter Ferris and A. R. Dodge discussed the paper, a written discussion by Prof. W. B. Gregory being read by the Secretary.

The paper on An Electric Gas Meter was presented by the author, Prof. Carl C. Thomas, of the University of Wisconsin. The paper describes a meter measuring the rate of flow of gas or air, which can be adapted for use as a steam meter or as a steam calorimeter. The operation of the gas meter depends upon the principle of adding electrically a known quantity of heat to the gas and determining the rate of flow by the rise in temperature of the gas (about 5 deg. fahr.) between inlet and outlet. The adoption of this principle of operation permits the construction of a very accurate and sensitive autographic meter of large capacity containing no moving parts in the gas passage; independent of fluctuations in pressure and temperature of the gas; and capable of measuring gas or air at either high or low pressures or temperatures. The electrical energy required is about 1 kw. per 50,000 cu. ft. hourly capacity, at the pressures ordinarily used in gas mains.

Prof. W. D. Ennis, E. D. Dreyfus and A. R. Dodge discussed the paper, a written discussion from Prof. L. S. Marks being read also.

THURSDAY AFTERNOON—STEAM ENGINEERING

At the Thursday afternoon session Vice-President L. P. Breckenridge presided. Five papers were presented dealing with different phases of steam engineering. The first paper, Tan Bark as a Boiler Fuel, by David M. Myers, described the results obtained by burning spent hemlock tan bark, the average fuel value of which is about 9500 B.t.u. per lb. of dry matter, which is about 35 per cent of its total moist weight in the fireroom. The available heat value per pound as fired is 2665 B.t.u. One ton of air-dry hemlock bark produces boiler fuel equal to 0.42 tons of 13,500 B.t.u. coal. A. A. Cary, Prof. Wm. Kent and Prof. L. P. Breckenridge took part in the discussion.

J. R. Bibbins then presented his paper on Cooling Towers for Steam and Gas-Power Plants, which contained a critical study of different types of towers with a description of their distinctive features. The paper also describes a simple inexpensive type of tower employing a lath-mat cooling surface and offers suggestions for a combination of natural-draft and forced-draft types.

The paper was discussed by Geo. J. Foran, W. D. Ennis, H. E. Longwell, B. H. Coffey, E. D. Dreyfus and F. J. Bryant. A written discussion by Carl G. de Laval was read by the Secretary.

W. P. Caine's paper, Governing Rolling Mill Engines, was read by Richard H. Rice. The paper describes and gives indicator cards and speed curves of a Corliss engine driving a three-high mill under two different conditions of governing, (*a*) under the widest range of adjustment of cut-off, (*b*) under a limited range, increasing the economy and making the engine run much more smoothly and safely. A table gives the power required for rolling in the mill and the momentary source of energy, whether from the cylinder or flywheel. A description is also given of the tachometer used to take the speed curves. Written discussions by H. C. Ord and James Tribe were read by the Secretary.

The next paper was that by F. W. Dean on An Experience with Leaky Vertical Fire-Tube Boilers. The author discussed the difficulties experienced with some large vertical boilers, somewhat over 10 ft. in diameter, and containing over 6000 sq. ft. of heating surface The boilers leaked badly very soon after being started and nothing that was done improved their condition until the water legs were lengthened from 2 ft. to 7 ft. $2\frac{3}{4}$ in., the boilers thus being raised 5 ft. $2\frac{3}{4}$ in. Before they were raised the lower ends of the tubes would cover with very hard clinker and become stopped up. This clinker could be removed only by cutting it off when the boilers were cold. After the boilers were raised, a light clinker that could be blown off formed about the tubes; by removing this by blowing every three or four hours the leaks were stopped and they have never returned.

Those taking part in the discussion were R. P. Bolton, Prof. Wm. Kent, J. C. Parker, O. C. Woolson, A. A. Cary, Prof. A. M. Greene, Jr., E. D. Meier and D. M. Myers. A. Bement submitted a written discussion.

Mr. Dean's second paper, The Best Form of Longitudinal Joint for Boilers, dealt with the defects of the usual form of butt joint used on the longitudinal seams of boilers, in which the inside strap is wider than the outside strap. It gave some history of the joint and discussed some of its defects and suggested a substitution for this form.

The paper was discussed by R. P. Bolton, Carl G. Barth, E. D. Meier, Prof. A. M. Greene, Jr., W. A. Jones, Prof. S. W. Robinson, Geo. I. Rockwood, and Sherwood F. Jeter.

GAS POWER SECTION

The session of the Gas Power Section was held on Thursday afternoon, Chairman F. R. Low presiding. In his address, the Chairman

referred briefly to the work of the various committees of the Section and stated that during the year the membership had increased from 247 to 378, a gain of over 50 per cent. Mr. Low also dealt with the development in the gas-power field during the year, mentioning some experiments with gas turbines. Gas-engine design, the use of by-product gases, the development of the bituminous producer, the gasification of peat, and the gas engine in marine work, were also briefly dealt with.

The report of the Tellers of Election, Edw. Van Winkle, Prof. Walter Rautenstrauch and J. V. V. Colwell, was then presented by Prof. Rautenstrauch, the results being as follows: for chairman J. R. Bibbins 107; for member of the Executive Committee, F. R. Low 108.

The report of the Gas Power Plant Operations Committee was then presented by James D. Andrew, and discussed by J. C. Parker, J. N. Norris and H. H. Suplee. Prof. C. H. Benjamin reported verbally for the Literature Committee, outlining the work of the committee in bringing gas-power literature to the attention of the members. H. R. Cobleigh and Professor Rautenstrauch also spoke on the work of this committee, the latter suggesting a plan for better organization of the committee to deal with literature on the subject.

L. B. Lent reported for the Gas Power Installations Committee that two forms had been prepared and sent to manufacturers, and while a good deal of information had been received, not enough was on hand for a complete report. The committee hoped to have the material in shape at an early date.

Prof. W. F. M. Goss then presented the paper on Testing Suction Gas Producers with a Koerting Ejector, by C. M. Garland and A. P. Kratz. The paper describes a method of testing the suction gas producer which is independent of the engine. The engine is blanked off from the producer and a Schutte & Koerting steam ejector is inserted, which draws the gases from the producer and delivers them to a scrubber in which the steam used by the ejector is condensed. The gases then pass to a meter for measuring their volume. Complete data of calculations and results are given in appendices.

The paper was discussed by Prof. R. H. Fernald, G. M. Tait, H. H. Suplee, L. B. Lent, S. C. Smith, W. B. Chapman and Edw. N. Trump.

The paper on Bituminous Gas Producers was then presented by the author, J. R. Bibbins. The paper describes a double-zone type of producer and the results obtained in gasifying bituminous coal. Continuous operation was secured with tar-free gas of reasonable heat value and producer efficiency and an over-all plant economy of about

one pound of fair bituminous coal per brake horsepower (proportionate economies for poorer grades). The efficiency and general effectiveness of operation of the producer on low-grade fuel, lignites, etc., was practically as high as with the higher grades. The following took part in the discussion: G. M. Tait, Prof. R. H. Fernald, W. B. Chapman, H. M. Latham, H. H. Suplee, Edw. N. Trump, H. B. Langer, S. C. Smith, Prof. W. Rautenstrauch, and G. D. Conlee.

FRIDAY MORNING

The session on Friday morning opened with the paper by Walter Ferris on The Bucyrus Locomotive Pile Driver. This paper describes a new railway pile driver, the leading feature of which is a very powerful propelling apparatus and a large boiler, enabling it to act as a locomotive and haul its own train of tool cars, boarding cars, etc., over the road. A special turn-table, consisting of hydraulic lifting apparatus and a large ball-bearing, enables the entire pile driver, including trucks, to be turned end for end or crosswise of the tracks. O. K. Harlan discussed the paper, A. F. Robinson and L. J. Hotchkiss submitting written discussions.

The paper by Henry Hess on Lineshaft Efficiency, Mechanical and Economic, described the test of the relative efficiency of a lineshaft of $2\frac{7}{16}$ in. diameter, making 214 r.p.m., with bearing load due to the weight of the parts plus the tension of the belts subjected to known stress by counterweighting, when running in ring-oiling babbitted bearings and when mounted in ball bearings. The savings in power consequent on this change ranged from 14 to 65 per cent, with 36 and 35 per cent under average conditions of good practice, due to belt tensions of 44 lb. and 57 lb. per inch width of single belt respectively. The paper gives data for determining the power savings that may be expected in various plants, by the use of ball bearings.

Those discussing the paper were T. F. Salter, Prof. R. C. Carpenter, C. A. Graves, O. K. Harlan, C. J. H. Woodbury, Walter Ferris, Fred J. Miller, A. C. Jackson, C. D. Parker and Oliver B. Zimmerman. Geo. N. Van Derhoff submitted a written discussion.

A. F. Nagle's paper on Pump Valves and Valve Areas, called the attention of engineers to the need of reviewing the common notion that "valve-seat area" is synonymous with "velocity of flow." The purpose of specifications for pumping engines is to secure a low velocity of flow through the valves, thus reducing the head required to force water through the pump; but to accomplish this purpose, special

and intelligent attention should be given to the springs of the valves, rather than to valve-seat areas. If that be done, valve-seat areas need not be greater than the plunger area for the vertical triple-expansion pumping engines so largely used in city pumps. Prof. W. M. Kent, A. B. Carhart, Prof. R. C. Carpenter and E. H. Foster discussed the paper. Contributed discussions were by Chas. A. Hague, I. H. Reynolds and F. W. Salmon.

Another paper by Mr. Nagle, a Report on Cast-Iron Test Bars, brought out the fact that test pieces, whether cast in separate molds or in the same mold as the main casting, are not perfect indications of the character of the iron in the main casting. The results obtained by the author would indicate a probable variation of 15 per cent where uniformity might be expected. A. A. Cary and T. M. Phetteplace discussed the paper, contributed discussion being by Prof. W. B. Gregory and Geo. M. Peek.

The meeting closed with the following resolutions, offered by Luther D. Burlingame:

Whereas The American Society of Mechanical Engineers at its Annual Meeting, December 1909, desires to express its appreciation to those who have provided opportunities for entertainment and on behalf of the visiting members and their guests thanks for the cordial welcome extended by the local members and their friends of New York and vicinity,

Be it ***Resolved*** that the Secretary extend the thanks of the Society and express the appreciation of its members and guests to the local committee for their untiring efforts, to those who have sent invitations to visit technical and engineering works and places of interest, to Mr. Geo. Gibbs, chief engineer of the Pennsylvania Tunnel and Terminal Railroad Co., and to Mr. Walter Kerr, president of the Westinghouse, Church, Kerr & Co., and their associates, for the opportunity to inspect the new Pennsylvania Railroad station; to Dr. B. T. Galloway, chief of the Bureau of Plant Industry, Department of Agriculture, for the very instructive and entertaining paper on The Era of Agricultural Machinery, and especially to those ladies who have so efficiently assisted by extending a generous hospitality to their guests.

EXCURSIONS

As usual at Conventions of the Society there were numerous excursions to points of interest in New York and vicinity, which constituted an important feature of the program for the entertainment

of visiting members and guests. Invitations for these excursions were generously extended by many firms and individuals, and through the efforts of the Excursion Committee, Hosea Webster, *Chairman*, trips to various plants and industries were arranged, to the representatives of which the grateful appreciation of the Society has been expressed.

A list of excursions follows:

Pennsylvania Railroad Terminal and Passenger Station: Invitation by George Gibbs, Chief Engineer, Pennsylvania Tunnel Terminal R. R. Co., and member of the Society; Henry R. Worthington Hydraulic Works, Harrison, N. J., by William Schwanhausser, Chief Consulting Engineer of International Steam Pump Co., member of the Society; Harrison Lamp Works of General Electric Co., Harrison, N. J., by George H. Morrison, General Manager; Interborough Rapid Transit Co., central power station at 59th St., New York, by H. G. Stott, Superintendent of Motive Power, Manager of the Society; Edison factories and Edison Laboratory at Orange, N. J., by Frank L. Dyer, President of National Phonograph Co., associate member of the Society; De La Vergne Machine Co., New York, by Adolf Bender, President; New York Telephone Co.; Gramercy and Stuyvesant Central Offices, by E. F. Sherwood, Superintendent of Traffic; Crocker-Wheeler Co., Ampere, N. J., by S. S. Wheeler, President, member of the Society; Westinghouse Lamp Co., Bloomfield, N. J., by Walter Carey, General Manager; New York Edison Co., Waterside Stations Nos. 1 and 2, by John W. Lieb, Jr., 3d Vice-President, member of the Society; Astoria Light, Heat & Power Co., Astoria, N. Y., by William H. Bradley, Chief Engineer, Consolidated Gas Co., member of the Society; Brooklyn Rapid Transit Co., Williamsburg Power Station, by C. E. Roehl, Electrical Engineer; Rockland Electric Co., Hillburn, N. Y.; Singer Building, New York, by Singer Mfg. Co.; Trenton Iron Co., Trenton, N. J.; Watson-Stillman Co., Ampere, N. J.; Metropolitan Life Insurance Building, New York.

Every possible courtesy was extended to the visiting parties in each case and in some instances special transportation facilities were provided. At the Edison Laboratory visitors were met by Thomas A. Edison, Hon.Mem.Am.Soc.M.E., who personally explained many points of interest about the plant. In order to avoid confusion, arrangements were made to assemble at the various manufactories at a time and place indicated in the program. The Information Bureau, located in the foyer of the building, under the chairmanship of F. E. Idell, was of material aid in this connection.

ENTERTAINMENT FEATURES

The Ladies' Reception Committee, composed of ladies resident in and about New York, under the chairmanship of Mrs. Herbert Gray Torrey, contributed much to the pleasure of members and guests of the Society. Tea was served from four until six o'clock on Tues-

day, Wednesday and Thursday afternoons during the convention, in the ladies' headquarters in the reception rooms of the Society on the eleventh floor. Mrs. George H. Westinghouse was the guest of the committee on Wednesday afternoon.

A number of excursions to shops and hotels were arranged and successfully carried out under the guidance of members of the committee. The kindness of Mr. and Mrs. John W. Lieb, Jr., made possible several enjoyable automobile rides through Central Park and Riverside Drive.

MEETINGS OF THE COUNCIL

DECEMBER 7, 1909

A meeting of the Council was called to order December 7, 1909, in the rooms of the Society, with President Smith in the chair. There were present at the meeting Geo. M. Basford, Geo. M. Bond, L. P. Breckenridge, R. C. Carpenter, H. L. Gantt, A. C. Humphreys, F. J. Miller, A. M. Waitt, Past-Presidents Charles Wallace Hunt, F. R. Hutton, Ambrose Swasey, F. W. Taylor and S. T. Wellman, and Calvin W. Rice, Secretary. The Council was especially pleased to have present John Fritz, Honorary Member and Past-President.

The minutes of the previous meeting were read and approved. The Secretary reported the deaths of Charles H. Willcox and William Metcalf.

The amendments to By-Laws B-6, B-7, B-12, B-13, B-18, B-19, B-27, B-28, B-34, and B-36 and the new By-Laws, one providing for the appointment of a Trustee of the United Engineering Society and one respecting The Journal of the Society, were approved.

EXECUTIVE COMMITTEE

Voted: That the Council sees no objection to any group of members selecting their own fiscal agent or correspondent, through whom the transmittal of their dues and other indebtedness to the Society may be made.

Voted: To refer the communication of the Western Society of Engineers, regarding the revision of the building laws of the State of Illinois, to the Public Relations Committee, to be appointed.

Voted: To approve the exchange of house and library privileges with the Louisiana Engineering Society.

The Secretary reported that circulars regarding the Joint Meeting in England had been issued to the membership and 116 favorable replies had already been received.

Voted: That the Council approve the recommendation of the Executive Committee approving coöperation with the Association of

American Steel Manufacturers to secure the general adoption of a system approved by a committee of the Society, December, 1894 (Trans., vol. 16, p. 32), to call the thickness of metals by their dimensions in decimals of an inch rather than by arbitrary number, and the Council recommends that the President appoint a committee to cöoperate with the Association.

The following were constituted such a committee: S. T. Wellman and George M. Bond.

The resolution referred to the Council from the Washington meeting, regarding the increase of facilities of the United States Patent Office, was laid on the table.

FINANCE COMMITTEE

The following resolutions were received from the Finance Committee and on motion approved:

That the Finance Committee recommend to the Council that the transfer of 10 per cent of the Reserve Fund of the Current Income Account be discontinued, as rceommended in the Annual Report.

That the Secretary be authorized to charge against this Annual Meeting Sub scription Fund, namely, $205.60, whatever bills may have been incurred by the office in behalf of the Local Committee for the Annual Meeting, and the balance, if any, be paid by the Treasurer to the Local Committee of 1909.

Voted: To approve the recommendation of the Finance Committee that a committee be appointed to take up the consideration of the question of increasing the membership and providing ways and means to put the same into effect during the coming year.

LIBRARY COMMITTEE

Voted: To adopt the following resolutions of the Library Committee but with the amendment that the House Committee have the first option on duplicate books, to enable that Committee to furnish the reception room:

To recommend to the Council that the Librarian be authorized to sell to the highest bidder, for the benefit of the Society, the duplicate books recommended in the letter of the Librarian to the Committee, dated July 23, 1909.

MEETINGS COMMITTEE

Voted: To receive the resolution of the Meetings Committee, to whom had been referred the action of the Council on a Machine Shop Section, but to amend to read:

Voted: To advise the Council that the Committee is in accord with the plans of the Council for carrying out the purpose of a Machine Shop Section through committees appointed by the Meetings Committee and not by the formation of a special section; and that the Committee will proceed to such plans as soon as possible.

Voted: To approve the recommendation of Atlantic City for the Spring Meeting of the Society, May 31 to June 3, 1910.

STUDENT BRANCHES

Voted: On recommendation of Professor Hutton, Chairman of the Sub-Committee on Student Branches, to approve the applications of the University of Nebraska at Lincoln, Neb., and the University of Missouri at Columbus, Mo., to form student branches of the Society.

A communication was read by the Secretary regarding the possibility of holding meetings of the Society in Chicago, along the lines of those in St. Louis and Boston.

The Secretary presented a draft of the annual report of the Council which after amendment, was approved and ordered filed and printed as the report of the Council for 1909.

Voted: That the Library Committee be requested to give consideration to the question of procuring and caring for a collection of lantern slides.

THURSTON MEMORIAL

Dr. Humphreys reported the intention of the Thurston Memorial Committee to have the dedication exercises in February at the regular monthly meeting of the Society, and requested suggestions from the Council of suitable speakers for that evening, covering the various phases of Dr. Thurston's life work at the Naval Academy, Stevens Institute and Cornell University, as well as his laboratory and research work and work in connection with the organization of the Society.

The meeting adjourned.

DECEMBER 10, 1909

A meeting of the Council was called to order by Jesse M. Smith, Past-President, on December 10, 1909, in the rooms of the Society.

Mr. Smith appointed Vice-President R. C. Carpenter and Manager I. E. Moultrop a committee to introduce to the Council the Vice-Presidents-elect and Managers-elect, and Past-Presidents Taylor and Hutton to introduce the President-elect, George Westinghouse.

Mr. Westinghouse then took the chair.

There were present at the meeting: President, George Westinghouse; Vice-Presidents, Chas. Whiting Baker, Geo. M. Bond, R. C. Carpenter, W. F. M. Goss, E. D. Meier, F. M. Whyte; Managers, J. Sellers Bancroft, H. L. Gantt, James Hartness, Alex. C. Humphreys, I. E. Moultrop, H. G. Reist, H. G. Stott; Past-Presidents, F. R. Hutton, Charles Wallace Hunt, Jesse M. Smith; Chairman Finance Committee, Arthur M. Waitt, and Secretary, Calvin W. Rice. Regrets were received from Treasurer, Wm. H. Wiley, and Manager, W. J. Sando.

The minutes of the meeting of December 7 were read and approved.

In the absence from the room of Calvin W. Rice, Secretary of the Society for the year 1909, H. G. Stott acted as Secretary *pro tem.*

Voted: That Calvin W. Rice be elected Secretary for the year 1910 on the same terms as the previous year.

Voted: That F. R. Hutton be elected Honorary Secretary for the year 1910, on the same terms as the previous year.

Voted: That Jesse M. Smith, Past-President, be elected Trustee of the United Engineering Society to serve for a term of three years, to fill the vacancy created by the expiration of the term of office of Charles Wallace Hunt.

Voted: That Henry R. Towne be reappointed a member of the John Fritz Medal Committee, under the provisions of C-46 and B-32 to serve for a term of four years, to succeed himself.

Voted: That the Council delegate to the President the appointment of the Executive Committee of the Council for the year 1910 and until the appointment of the new Executive Committee the present Executive Committee continue in service.

The meeting adjourned to January 11, 1910.

THE NEWLY ELECTED OFFICERS FOR 1910

GEORGE WESTINGHOUSE

President Am. Soc. M. E.

George Westinghouse, a son of George and Emeline Vedder Westinghouse, was born at Central Bridge, N. Y., October 6, 1846. His father was a manufacturer of agricultural machinery, and established works at Schenectady, which are still in operation. The younger Westinghouse was educated in the public schools and at Union College, Schenectady, and received his early mechanical training in his father's manufactory. His tastes were strongly in the direction of machinery and the solution of mechanical problems.

The patriotic ardor which filled the youth of the country during the civil war drew young Westinghouse into the volunteer army in June 1863. He was under seventeen, but on account of his size and strength—he was six feet tall and weighed 180 lb.—the recruiting officers admitted him without asking his age. He enlisted with the Twelfth New York National Guard. Subsequently, he joined the Sixteenth New York Cavalry, and in December 1864 became an assistant engineer in the United States Navy, serving in that capacity until August 1865.

Returning to civil life he invented in the same year a device for replacing derailed cars, and while placing this invention with the railroads his attention was attracted by the prevalence of minor and serious accidents due to the lack of efficient means for controlling trains in motion. After a careful study of the subject, and such experiments as were possible with the limited means then obtainable, he invented the air brake and patented it in 1868.

The first train to which this brake was applied ran on a line west from Pittsburgh and on what is now a portion of the Pennsylvania Railroad. During the trial trip a collision with a loaded team stuck on a grade-crossing was prevented. This practical illustration of the utility of the invention led to the adoption of the brake. Mr. Westinghouse, retaining the control of his invention, undertook to manufacture it and organized the Westinghouse Air Brake Company,

establishing at Pittsburgh the business which subsequently became the nucleus of the many industries associated with his name.

From the invention of the air brake dates the beginning of modern railroading. The air brake is primarily a train-operating device which makes possible the fast and long trains, large cars, heavy loads and frequency of service of the present day, and the numerous improvements which Mr. Westinghouse has wrought in his invention have kept its efficiency well in advance of the new and varied conditions which constantly arise. Before he was twenty-five his name had become familiar throughout the world, and his contribution to the material progress of civilization was everywhere recognized. He continued in the study and practice of engineering, and equipped a machine shop for his personal experimental use, where he worked out many inventions, at first relating almost entirely to devices for railroad operations. He applied compressed air to switching and signalling and later utilized electricity in this connection. From this grew the Union Switch and Signal Co.

His introduction of electricity into switch and signal work led him far into electrical experiment and he devoted his energies to a cause in which few then believed, the adoption of the alternating current for lighting and power, in which he had to meet and overcome almost fanatical opposition, which in many States sought legislation against the use of the alternating current as dangerous to the public welfare. In 1885 he acquired the patents of Gaulard & Gibbs, and having undertaken a comprehensive study of the distribution and utilization of electrical currents in a large way, he personally devised apparatus and methods for the work, and gathered around him a group of men who were to become experts in the new electrical art. He also organized the electrical company which bears his name and undertook the development and manufacture of the induction motor which made practical the utilization of the alternating current for power purposes.

Following the discovery of natural gas in the Pittsburgh region, Mr. Westinghouse devised a system for controlling the flow and for conveying the gas over long distances through pipe lines, thus supplying fuel to the homes and factories of Pittsburgh. He took up the study of the gas engine, and for ten years conducted a series of exhaustive experiments in this line, at the end of that time putting into commercial use a gas engine of large power for electric generating.

Mr. Westinghouse introduced the Parsons steam-turbine into this country, adding to it improvements and developments of his own,

and others carried out under his supervision. He also has recently developed a steam turbine for ship-propulsion designed to overcome the well-known objections to the use of turbines in that field, and lately coöperated with Rear-Admiral Melville and John H. Macalpine in their study of problems associated with driving of propellers at low speed by turbines of high speed.

It is impracticable to enumerate here the inventions which Mr. Westinghouse has personally made or those which his staff have brought forth under his supervision. As a result of this work and enterprise, there have grown thirty corporations of which he is president, employing 50,000 men, $120,000,000 of capital, with works at Wilmerding, East Pittsburgh, Swissvale and Trafford City, Pa.; at Hamilton, Canada; London and Manchester, England; Havre, France Vardo, Italy; and at Vienna and St. Petersburg.

Mr. Westinghouse has made many visits to Europe in connection with his inventions and industries. There as in his own country he has won the friendship of the foremost men of his time and the high esteem of the engineering profession. He has been decorated by the French Republic and by the sovereigns of Italy and Belgium; and he was the second recipient of the John Fritz Medal, Lord Kelvin, his friend of many years, having been the first. The Königliche Technische Hochschule of Berlin bestowed upon him the degree of Doctor of Engineering; and his own college, Union, gave him the degree of Ph.D. In 1905 Mr. Westinghouse was selected as one of the three trustees in whose hands the voting power of the controlling stock interest in the Equitable Life Insurance Society was placed. The other trustees were Ex-President Grover Cleveland, and Justice Morgan J. O'Brien. The selection of these three men met with universal approbation. Besides his Honorary membership in The American Society of Mechanical Engineers, Mr. Westinghouse is one of the two honorary members of the American Association for the Advancement of Science and is an honorary member of the National Electric Light Association.

Mr. Westinghouse married, in 1867, Miss Marguerite Erskine Walker and has one son, George Westinghouse, Jr. While he claims Pittsburgh as his residence, he has also a country home at Erskine Park, Lenox, Mass., as well as a house in Washington.

VICE-PRESIDENTS

CHARLES WHITING BAKER

Charles Whiting Baker, editor and vice-president of Engineering News, was born in Johnson, Vt., January 17, 1865, and was educated at the State Normal School at Johnson and at the University of Vermont. He received the degree of Civil Engineer from the latter institution in 1886. During his course Mr. Baker spent one vacation as aid on triangulation work for the United States Coast and Geodetic Survey in Vermont, and on graduation he worked for a few months in the drafting room of the Baldwin Locomotive Works, at Philadelphia, Pa.

Leaving this position in February 1889 to become associate editor of Engineering News, of New York, Mr. Baker took up a work which has claimed his attention ever since. Since 1892 he has been in practical charge of the editorial department, becoming in 1895, on the death of A. M. Wellington, managing editor and secretary of the company. Ten years later he became vice-president.

Mr. Baker published in 1889 an economic work, Monopolies and the People, and he has contributed to the Society a paper entitled, What is the Heating Surface of a Steam Boiler, presented in June 1898. He joined the Society in 1893, and served on the Meetings Committee from 1905 to 1908.

WILLIAM FREEMAN MYRICK GOSS

William Freeman Myrick Goss was born in Barnstable, Mass., October 7, 1859. In 1879 he received the certificate of the Massachusetts Institute of Technology, and afterwards the degree of Hon. M.S., from Wabash in 1888, and D.Eng., from the University of Illinois in 1904.

In 1879 Dr. Goss became an instructor in the department of mechanic arts of Purdue University, and remained in the service of that institution for nearly thirty years, becoming successively professor of practical mechanics in 1883 and professor of experimental engineering in 1889. He was made a director of the engineering laboratory in 1899, and dean of the school of engineering in 1900. In 1907 Dr. Goss entered the University of Illinois as dean of the college of engineering and director of the school of railway engineering and administration.

Dr. Goss served on the jury of awards for the Columbian Exposition in 1893, and has been a member since 1906 of the executive committee of the National Advisory Board on Fuels and Structural Materials. He is a fellow of the American Association for the Advancement of Science, and a member of the Society for the Promotion of Engineering Education, the Western Railway Club, the Western Society of Engineers, the American Institute of Electrical Engineers, the Illinois Academy of Science, the Master Car Builders' Association, the Master Mechanics' Association, the International Association for Testing Materials, and the Illinois Society of Engineers and Surveyors.

Dr. Goss has made a specialty of the subject of steam engineering, investigating largely the economic performance of locomotives, high pressure in locomotive service, superheated steam in locomotive service, behavior of car axles, friction brakes, front-end arrangement of locomotives, fuel briquets in locomotive service, power transmission by friction wheels, graphite as a lubricant, etc.

Dr. Goss is a life member of this Society, which he entered in 1886. He was a member of the board of managers from 1900 to 1903, and has served on many committees. He has contributed the following papers: The Cole Locomotive Superheater; A Series Distilling Apparatus of High Efficiency; The Effect of the Counterbalance in Locomotive Drive Wheels upon the Pressure between Wheel and Rail; Tests of a Ten-Horsepower DeLaval Steam Turbine; New Forms of Friction Brakes; Tests of the Locomotive at the Laboratory of Purdue University; Paper Friction Wheels; Tests of a Twelve-Horsepower Gas Engine; Efficiency Tests of a One Hundred Twenty-Five Horsepower Gas Engine; The Effect upon the Diagrams, of Long Pipe Connections for Steam-Engine Indicators; Test of the Snow Pumping Engine at the Riverside Station of the Indianapolis Water Company; Locomotive Testing Plants; Power Transmission by Friction Driving; The Conservation of the Nation's Fuel Supply; The Debt of Modern Civilization to the Steam Engine.

EDWARD DANIEL MEIER

Colonel Edward Daniel Meier, president and chief engineer of the Heine Safety Boiler Conpamy, was born in St. Louis, Mo., May 30, 1841. At the close of a scientific course at Washington University, St. Louis, he studied four years at the Royal Polytechnic College at Hanover, from 1859 to 1862. He was then apprenticed to Wm. Mason's

Locomotive Works at Taunton, N. J. He left this company for military service, part of the time doing construction work as assistant engineer on the defenses of New Orleans.

In 1865 Colonel Meier entered the Rogers Locomotive Works at Paterson, N. J., as machinist and draftsman, During the next ten years he held various important positions with the Kansas Pacific Railway, the Illinois Patent Coke Company, the Meier Iron Company, the St. Louis Interstate Fair, and the St. Louis Cotton Factory. From 1876 to 1879 he was designer and superintendent of the Peper Hydraulic Cotton Press, and after two years of varied administrative work became president and chief engineer of the Heine Safety Boiler Company, which offices he still holds. During that period he has acted as consulting engineer on the Union Depot Railway of St. Louis, constructing the first electric power station in that city; and from 1902 to 1908 as engineer-in-chief and treasurer of the American Diesel Engine Company.

Colonel Meier has held office in the St. Louis Engineers' Club, the American Boiler Manufacturers' Association, and the Machinery and Metal Trades Association. He entered this Society in 1891 and has served it as manager, from 1895 to 1898, and as vice-president from 1898 to 1900.

MANAGERS

J. SELLERS BANCROFT

Mr. J. Sellers Bancroft was born September 12, 1843, and was educated in the public schools of Philadelphia.

In March 1861 he was apprenticed to the machinery business with Wm. Sellers & Co., with whom he was advanced to gang foreman in 1863, before the completion of his apprenticeship, and shop foreman in 1867, becoming a member of the firm in 1873, and manager of the business from its incorporation in 1887 to January 31, 1902, when he left this company to become general manager and mechanical engineer for the Lanston Monotype Machine Company, builders of monotype-casting and composing machinery. Mr. Bancroft has taken out over sixty patents for various inventions in machine tools, injectors, testing machines, electrical appliances, and type-casting and composing machines, and is largely responsible for the present condition of monotype machinery. He received a gold medal from the Paris Exposition of 1889 for his inventions in machine tools and injectors there shown.

Mr. Bancroft is a member of the American Association for the Advancement of Science, and has been a member of the Franklin Institute for over forty years. He entered this Society in 1880.

JAMES HARTNESS

James Hartness was born in Schenectady, N. Y., September 3, 1861, and received his early training in the public schools. After seven years of experience as machinist, toolmaker and draftsman, he became foreman and designer for the Union Hardware Company, of Torrington, Conn., a position which he relinquished after three years to become superintendent and designer for the Jones & Lamson Machine Co., of Springfield, Vt. He has had an active part in the management of this firm for nearly twenty-one years, and has been its president for the last nine years. Mr. Hartness is also president of the Bryant Chucking Grinder Company, treasurer of the Jones & Lamson Power Co., and director in a number of other machine tool building companies.

He has taken out seventy United States patents, besides many pending, his line of invention being machines for metal turning, notably the flat turret lathe.

Mr. Hartness published in 1909 a work on Machine Building for Profit and the Flat Turret Lathe, and has contributed to the Society, papers on Lead-Controlling Screw-Cutting Dies, and Tandem Dies, in 1897, and Metal-Cutting Tools without Clearance, in 1908. He joined the Society in 1891 and is a life member. He is also a member of the Institution of Mechanical Engineers of Great Britain, the American Society for the Advancement of Science, the American Institute for Scientific Research, and the Boston Chamber of Commerce, as well as the Engineers' Club, and various other social organizations.

HENRY G. REIST

Henry G. Reist was born near Mt. Joy, Lancaster County, Pa., May 27, 1862. He received from Lehigh University in 1886 the degree of M.E. The same year he entered the foundry and machine department of the Harrisburg Car Company. After a year of testing and erecting steam engines he became assistant superintendent of the company.

Leaving in the spring of 1889 to join the engineering excursion to Europe, he became associated on his return with the Thomson-

Houston Electric Company, at Lynn, Mass., having charge of the construction and testing of a large number of direct and alternating-current dynamos and stationary and railway motors. Soon after the consolidation of the Thomson-Houston Company with the General Electric Company, he took charge for them of the design of alternating-current generators and motors. When he was first engaged in electrical work, the largest machine manufactured by the company with which he was associated, was of 100-kw. capacity; now 14,000-kw. generators are regularly produced by this company.

Mr. Reist entered this Society in 1889 and somewhat later became a member of the American Institute of Electrical Engineers. He has contributed to the Society a paper on Blueprinting by Electric Light, and has presented papers before the American Institute of Electrical Engineers, and the Ohio Electric Lighting Association of Engine Builders, as well as a number of lectures to engineering students.

GENERAL NOTES

STUDENT BRANCHES

The following reports have come to the Society concerning the activities of its Student Branches:

At Columbia University the following officers were elected recently: F. R. Davis, president, H. B. Egbert, vice-president, H. B. Jenkins, secretary, and F. T. Lacy, treasurer. Papers are read before the organization once a month.

At Brooklyn Polytechnic a number of new members were received at the meeting of December 4, and a lecture was delivered by H. A. Black, on Depreciation Principles and Methods.

The Mechanical Engineering Society of the Massachusetts Institute of Technology enjoyed a lecture on December 21 by Robert A. Shailer, on Tunnels and Tunnel Construction. The society conducts excursions from time to time to places of industrial interest, those lately visited being the Quincy Market Coal Storage and Warehouse Co.'s refrigerating plant and the factory of the Stanley Motor Carriage Company. The officers are Frederick A. Dewey, chairman, Donald V. Williamson, vice-chairman, Arthur P. Truette, secretary, and Luke E. Sawyer, treasurer.

On December 3, the recently organized branch at the University of Cincinnati elected as temporary officers H. B. Cook, chairman, and P. G. Haines, secretary.

The Club of Mechanical Engineering of the University of Missouri, which was admitted at the last meeting of the Council on December 7 as a student branch of the Society, elected R. E. Dudley, president, Ernest C. Phillips, secretary-treasurer, and for members of the advisory board, Prof. E. A. Fessenden, Jun., Am.Soc.M.E., E. C. Phillips, and F. B. Thatcher. The club has as its honorary chairman Prof. Harry Wade Hibbard, Mem. Am.Soc.M.E.

Further statistics concerning these and other student branches are published on another page of The Journal.

ENGINEERS' CLUB BANQUET

The third annual banquet of the Engineers' Club took place Wednesday evening, December 22, with Mr. Andrew Carnegie, Honorary

Member Am.Soc.M.E., as the guest of honor. Mr. Carnegie mentioned his great pleasure in attending the dinner, an attendance which he regarded in the light of an obligation, and spoke again of his debt to the engineers and the chemists for their part in all his industrial success. He also repeated his prophecy that in the course of time Canada and the United States would be one nation.

This remark served to introduce another guest of the evening, Robert Cooper Smith, Esq. K.C., of Montreal, Quebec. Mr. Smith's address was an eloquent tribute to Mr. Carnegie. Mr. Martin W. Littleton followed and, in the absence of Hon. E. H. Gary, the speeches of the evening were concluded by Dr. Alex. C. Humphreys. Dr. Humphreys referred to the value of industrial education such as Mr. Carnegie is so successfully providing in the Carnegie Technical Schools at Pittsburgh, and reiterated his opinion that the educational work in America is too much influenced by the college, instead of training the average person for industrial life.

The attendance was nearly 200 and the excellence of the speeches and of the music, rendered by an orchestra under the direction of Hans Kronold, and the perfection of the menu and service made the occasion one of the most enjoyable and successful ever held by the Club.

DONATION TO THE LIBRARY

Clarence E. Kinne, Life Member, Am.Soc.M.E., in response to a request sent out through The Journal, has made up from his own files and sent to the Society the copies for 1894-1895, complete with index, forming vol. 1 of Machinery, which the Society had been unable to obtain through the customary channels. The volume has been placed in the Library, completing our files of this magazine to date.

REPRESENTATION AT FUNERAL SERVICES OF HORACE SEE

The President appointed James M. Dodge, Past-President, Rear Admiral George W. Melville, Past-President and Honorary Member, Oberlin Smith, Past-President, Fred. W. Taylor. Past-President, and J. Sellers Bancroft, Kern Dodge and Edward I. H. Howell, Honorary Vice-Presidents to represent the Society at the funeral services of Horace See, Past-President, Am.Soc.M.E., Thursday, December 16, 1909, at St. Peter's Church, 4th and Pine Sts., Philadelphia, Pa.

OTHER SOCIETIES

AMERICAN EXPOSITION IN BERLIN

As the first all-American Exposition ever conducted in a foreign country, the exposition to be held in Berlin during the summer of 1910 will offer peculiar advantages to American manufacturers. A freight reduction of 30 per cent both ways, granted by the Hamburg-American and the North German Lloyd lines, the remission of customs duty by the German Government, and the existence of the German-American patent treaty, which relieves American inventors from the necessity of obtaining patents in Germany, are among the inducements offered to exhibitors. The date set for the opening is June 20, and the exposition will be in progress three months. The exhibits will be carefully classified, the present plans including sections to be devoted to inventions, transportation, social economy and industrial safety, agricultural implements, machinery of all kinds, etc. Germany in 1908 consumed American products to the amount of $276,922,089; to say nothing of her influence on the trade of Europe.

The American headquarters for the exposition are in the Hudson Terminal Building, 50 Church St., New York, James L. Farmer, General Secretary. Members of the Society acting on the advisory committee are, C. A. Moore, Francis H. Stillman, Ambrose Swasey; and on the general committee, James M. Dodge and Thomas A. Edison, Honorary Member.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

At a meeting of the American Institute of Electrical Engineers held in the auditorium of the Engineering Societies' building, December 16, a paper entitled Comments on Development and Operation of Hydroelectric Plants, was presented by Henry L. Doherty, Member Am.Soc.M.E. The meeting was under the auspices of the High-Tension Transmission Committee.

WESTERN SOCIETY OF ENGINEERS

The Western Society of Engineers has appointed the following as a committee to confer with the Chicago City Council and the Harbor Commissioner regarding harbor improvement and development: A. Bement, Mem.Am.Soc.M.E., *Chairman*, W. L. Abbott, Member of Council, Am.Soc.M.E., L. E. Ritter, E. C. Shankland, Mem.Am.Soc.M.E., Willard A. Smith.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

The annual meeting of the American Institute of Chemical Engineers was held at Philadelphia, Pa., December 8 to 10. The address of welcome was made by Mayor John E. Reyburn. The following papers were presented for discussion: Natural Draft Gas Producers and Gas Furnaces, Ernest Schmatolla; The Commercial Extraction of Grease and Oils, W. M. Booth; The Chemical Industries of America, Prof. Chas. E. Munroe; Multiple Effect Distillation, F. J. Wood, Mem. Am.Soc.M.E.; The Advantages of the Multiple Effect Distillation of Glycerine and Other Products, A. C. Langmuir; Reclaiming of Waste India Rubber, S. P. Sharples; Materials for Textile Chemical Machines, Fred. Dannerth; A Method for Smelting Iron Ore in the Electric Furnace, Edw. R. Taylor; Chemical Composition of Illinois Coal, and Heat Efficiency of Smokeless Combustion and Heat Absorbing Capacity of Boilers, A. Bement, Mem.Am.Soc.-M.E.

Excursions were made to the laboratories of the University of Pennsylvania and the Commercial Museum, the chemical works of Harrison Bros. & Co., the Torresdale Filtration Plant; the wool-degreasing plant of Erben, Harding & Co.; the Welsbach Light Company, the plant of the Camden Coke Company; the Trenton Potteries; the Hamilton Rubber Company; the Linoleum Works; the cement plant at Allentown, Pa.

NATIONAL SOCIETY FOR THE PROMOTION OF INDUSTRIAL EDUCATION

The National Society for the Promotion of Industrial Education held its third annual convention at Milwaukee, December 2–4. The convention was opened with a public banquet at the Hotel Pfister, at which James O. Davidson, Governor of Wisconsin, presided. Addresses on the Economic Value of Industrial Education were made by Charles Van Hise, President of the University of Wisconsin, George

Martin, former secretary of the Massachusetts Board of Education, and Alex. C. Humphreys, Manager Am.Soc.M.E., President of Stevens Institute of Technology. Mr. Humphreys spoke particularly of the improvidence and superficial character of our educational processes which have built up a system that has the college as its goal, whereas in reality the masses need industrial training. The many are being sacrificed to the few.

Public meetings were held on the remaining days, at which National Legislation, Corporation Schools, Evening Schools, Industrial Education at Home and Abroad, and Intermediate Industrial Schools, were considered and discussed. Among those who addressed the gatherings were Willet N. Hayes, Assistant Secretary of Agriculture, John L. Shearer, President Ohio Mechanics' Institute, Arthur L. Williston, Mem.Am.Soc.M.E., Director in Pratt Institute, Mrs. Anna Garlin Spencer, Society for Ethical Culture, and Edgar S. Barney, Superintendent Hebrew Technical School for Boys. An exhibition of trade school work was conducted throughout the convention, some thirty prominent industrial institutions being represented.

The object of the society is to bring to public attention and to provide opportunities for the study of industrial education, as well as to make available the results of experience and to promote the establishment of additional institutions. Its work is carried on through a general office in New York and through State branches and committees. The New York State Branch has as its president James F. McElroy, Mem.Am. Soc.M.E., and as its secretary Prof. Arthur L. Williston, Mem.Am.Soc.M.E.

NATIONAL COMMERCIAL GAS ASSOCIATION

The fourth annual meeting of the National Commercial Gas Association occupied Madison Square Garden from December 14 to 22. One of the greatest undertakings of the exhibition committee was the piping of the entire building, making possible the most extensive and successful gas and gas appliance exhibition ever held. Among papers presented were: The Future of Gas for Street Lighting, E. N. Wrightington; The Use of Gas for Industrial Purposes, Present and Future, S. T. Wilson; The Application of Architectural Designs to Gas Fixtures, L. F. Blyler; Theory of Combustion, T. O. Horton; Gas Engines in Competition with Central Station Electric and Isolated Steam Plants, W. W. Cummings, Mem.Am.Soc. M.E.; General Maintenance and Special Troubles, R. H. Thomas;

Water Heaters, G. W. Savage. On December 17 a joint meeting of the Association with the New York Section of the Illuminating Engineering Society was held.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

The seventeenth annual meeting of the Society of Naval Architects and Marine Engineers was held at the Engineering Societies Building, New York on November 18 and 19. Among the papers presented for discussion were the following: The Foreign Trade Merchant Marine of the United States; Can it Be Revived? by G. W. Dickie, Mem.Am.Soc.M.E.; the Evolution of Screw Propulsion in the United States, by Chas. H. Cramp; The Effect of Parallel Middle Body upon Resistance, by D. W. Taylor; The Applications of Electricity to the Propulsion of Naval Vessels, by W. L. R. Emmet, Mem.Am.Soc.M.E.; The Strength of Water-tight Bulkheads, by Prof. William Hovgaard; The Design of Submarines, by M. F. Hay.

The officers elected are: president, Stevenson Taylor, Mem.Am. Soc.M.E.; vice-presidents, J. W. Miller, Rear-Adm. Geo. W. Melville, Hon. Mem.Am.Soc.M.E.; Members of Council, Wm. J. Baxter, Geo. W. Dickie, Mem.Am.Soc,M.E.,W. D. Forbes, Mem.Am.Soc. M.E., Andrew Fletcher, Mem.Am.Soc.M.E., H. A. Magoun, Mem. Am.Soc.M.E., Lewis Nixon; Associate Members of Council, J. S. Hyde, Assoc.Am.Soc.M.E., C. B. Orcutt.

ENGINEERS' CLUB OF ST. LOUIS

At the annual meeting of the Engineers' Club of St. Louis, held in the club rooms on December 1, 1909, reports covering the work of the year were presented, and the following officers placed in nomination and ordered to ballot: President, M. L. Holman, Past-President, Am.Soc.M.E.; vice-president, J. D. Von Maur; secretary-librarian, A. S. Langsdorf; treasurer, C. M. Talbert; directors, J. W. Woermann and H. J. Pfeifer; members of the board of managers, Association of Engineering Societies, John Hunter, Montgomery Schuyler, J. F. Bratney.

The business of the evening was followed by an illustrated address on Reinforced-Concrete Construction by A. J. Widmer, of the Trussed Concrete Steel Company.

BROOKLYN ENGINEERS' CLUB

The Brooklyn Engineers' Club held its annual meeting in the club-house, 117 Remsen St., on December 9. The annual reports of the various committees and the Board of Directors were read. Upon motion it was voted that the election of the new board of officers take place during the annual dinner, Thursday, December 16, at which time the following were elected: President, George A. Orrok, Mem.Am.Soc.M.E.; secretary, Joseph Strachan; treasurer, William T. Donnelly, Mem.Am.Soc.M.E.; directors, William Andrews, Frederick C. Noble; auditing committee for one year, Fred. L. Cranford, Jacob Schmitt, Geo. A. Hartung. During the afternoon and evening of the same day, the first annual loan exhibition of the scientific books of the year, photographs and plans of engineering work was opened.

NECROLOGY

HORACE SEE, PAST-PRESIDENT, AM.SOC.M.E.

The sudden death of Horace See, Past-President, Am.Soc.M.E., December 14, 1909, is announced. An account of his life will appear in an early number of The Journal.

The death of Dr. Charles B. Dudley, member of the Research Committee of the Society, December 21, 1909, is announced. An account of his life will appear later.

CHARLES HENRY WILLCOX

Charles Henry Willcox died at his home in Westport, Conn., on September 13, 1909. Mr. Willcox was born in Little Falls, N. Y., on March 31, 1839, and was the son of James Willcox, founder and president of the Willcox & Gibbs Sewing Machine Co. He entered his father's business at the age of eighteen and was continuously connected with the company as mechanical engineer from 1866 until his retirement a few years ago, and most of that time as director. The natural bent of his mind was toward mechanics and in collaboration with James E. A. Gibbs he developed and placed on the market the invention of the single-thread chain-stitch sewing machine, which is now so widely used in the making of wearing apparel. Other patents followed, in particular that of the automatic tension, which it is said consumed ten years of patient experimentation before it was perfected. Mr. Willcox was also the inventor of two straw-hat sewing-machines, one the American straw hat machine in which the stitch is visible, and another in which the stitch is concealed. These two machines are used to-day in the manufacture of fully 90 per cent of all straw hats made. The knit goods manufacturing field also received an impetus through the invention of the Willcox & Gibbs hosiery trimming machine. The overlock machine worked out by Mr. Willcox in collaboration with the late Stockton Borton, was a great advance over the hosiery trimming machine and is recognized as one of the finest mechanical productions

in sewing machines. Through the ornamental character of its stitch it has been adopted in lines of manufacture other than that for which it was originally intended.

In addition to his connection with the Willcox & Gibbs Sewing Machine Co., Mr. Willcox was for forty years affiliated with the Brown & Sharpe Mfg. Co. He was a life member of the Society.

CHARLES SWINSCOE

Charles Swinscoe, consulting engineer of the Clinton Wire Cloth Company, Clinton, Mass., was born at Nottingham, England, January 1, 1833. His early education was received in the Collegiate School, Manchester, England. He came to this country when a lad and at one time was Fourth Officer on the Dreadnought under Capt. Samuel Samuels.

From 1851 to 1854 Mr. Swinscoe studied practical mechanics in his father's shop. In 1867 he established the steam pump works of the Geo. F. Blake Mfg. Co., at Boston, designing most of the work. In 1876 he left this company to take charge of the Reading Hydraulic Works, designing its steam pumping machinery. From 1878 to 1880 he was in charge of the Bay State Brick Company and after that date of the Clinton Wire Cloth Company. In 1903 he became consulting engineer of this company.

Mr. Swinscoe was a musician of ability and was president of the Clinton Choral Union and organist of the Episcopal Church for many years. He was a member of the Clinton Historical Society, and a member of this Society since 1887.

PERSONALS OF THE MEMBERSHIP, AM. SOC. M. E.

Ludwell B. Alexander has assumed the position of vice-president of the Haggerty Contracting Company, Bronx Borough, New York. He was formerly associated with the United Engineering and Constructing Company, New York, as assistant engineer.

Thomas Appleton, formerly connected with the East St. Louis, Ill., office of the U. S. Public Buildings, as superintendent of construction, is now identified with the Alton, Ill., office.

Adolph O. Austin, chief draftsman of the Starr Engineering Co., New York, has accepted a position with the Vilter Mfg. Co., Milwaukee, Wis., in the capacity of assistant engineer.

C. Kemble Baldwin, formerly chief engineer of the Robins Conveying Belt Company, and for the past two years chief engineer of the Robins New Conveyor Company, has been appointed chief engineer of the Robins Conveying Belt Company, the two companies having been consolidated. Mr. Kemble lectured on The Belt Conveyor, on November 10, before 400 members of the first class of the engineering course of the University of Illinois.

A. Bement presented papers on Chemical Composition of Illinois Coal, and Heat Efficiency of Smokeless Combustion and Heat Absorbing Capacity of Boilers, at the December 8–10 convention of the American Institute of Chemical Engineers, held in Philadelphia, Pa.

Paul P. Bird presented a paper on The Smoke Problem of Chicago at the November 17 meeting of the Western Society of Engineers.

Walter J. Bitterlich, formerly machine designer with the Bresnahan Shoe Machinery Company, Lynn, Mass., has accepted a position with the Hood Rubber Company, Watertown, Mass., to act in the capacity of chief draftsman.

Paul M. Chamberlain has resigned the position of chief engineer of the Underfeed Stoker Company of America, Chicago, Ill., to take up private practice. His office will be in the Marquette building, Chicago, Ill.

Chas. C. Christensen contributed an article on A One Hundred Ton Modern Cyanide Plant to the November 13 issue of *The Mining World*.

H. V. Conrad has accepted a position with the Westinghouse Air Brake Company, Wilmerding, Pa.

George L. Crook, recently in charge of the manufacturing organization in the E-M-F plant at Detroit, Mich., has entered the employ of the M. Rumely Co., La Porte, Ind., as works manager.

Henry L. Doherty presented a paper entitled, Comments on the Development and Operation of Hydro-Electric Plants, at the December 16 meeting of the American Institute of Electrical Engineers.

Carl S. Dow contributed an article on The Fuel Economizer to the December issue of *The Practical Engineer.*

Frank B. Gilbreth is the author of a book on Bricklaying System.

Charles A. Hague delivered a lecture on The Development of the Pumping Engine, at the Sheffield Scientific School, Yale University, New Haven, Conn., November 12.

An article on Errors in Grinding Tapered Reamers and Milling Cutters, by H. A. S. Howarth, was published in the December number of *Machinery.*

Prof. Fred. R. Hutton delivered a lecture, on November 9, on Some Problems of the Large Gas Engine, before the Stevens Institute Engineering Society, affiliated with The American Society of Mechanical Engineers. Professor Hutton has been invited to give a lecture before the Graduate School of Marine Engineering, U. S. Naval Academy, Annapolis, in January.

A. Lewis Jenkins has contributed an article on Stresses due to Bending and Twisting and the Design of Shafting, to the November 12 issue of *Engineering* (London).

Charles Kirchhoff, who has been connected for almost thirty years with *The Iron Age,* and the other publications of the David Williams Company, has disposed of his interests in that company and retired from active business.

George L. Knight delivered a lecture on The Generating and Distributing System of the Brooklyn Edison Company, at the November 6 meeting of the Brooklyn Polytechnic Student Branch of the Society.

Prof. A. G. Koenig delivered an illustrated lecture on Refrigeration before the December 7 meeting of the Modern Science Club.

J. W. Lieb, Jr., delivered a lecture, December 10, before the Electrical Engineering Society of Columbia University, the subject being Electric Light.

John McGeorge and H. W. Woodward have formed a consulting firm under the name of the Cleveland Engineering Company, with offices in the New England Building, Cleveland. Mr. McGeorge has been chief engineer of the Wellman-Seaver-Morgan Co., Cleveland, O.

A biographical sketch of Spencer Miller was published in the December issue of *Cassier's Magazine.*

David M. Myers contributed an article on Burning Natural Gas as Boiler Fuel to the December 7 issue of *Power and the Engineer*.

E. W. Nicklin, recently identified with the Diamond Power Specialty Company, Detroit, Mich., has accepted a position with the Detroit Brass Works, Detroit, Mich.

George A. Orrok lectured before the Student Section of The American Society of Mechanical Engineers at Columbia University on the evening of December 3 on Gas Engine and Blast Furnace Practice. At the December 21 meeting of the Modern Science Club, Mr. Orrok delivered a lecture on Surface Condensers. Mr. Orrok has been elected president of the Brooklyn Engineers' Club.

Thos. C. Pulman, formerly manager in India for the Worthington Pump Company, Ltd., and James Simpson & Co., Ltd., subsidiary companies of the International Pump Co., of New York, has been appointed to the London offices of the companies, to supervise the Indian and Eastern business, and will be attached to the sales department.

R. H. Rice addressed a joint meeting of the Electrical Section of the Western Society of Engineers and the Chicago Branch of the American Institute of Electrical Engineers, December 22, on Low Tension Feeder Systems for Street Railways.

Morris DeF. Sample, formerly manager of department, National Patent Holding Company, Chicago, Ill., has become associated with The Fire Protection Company, Indianapolis, Ind., as secretary-treasurer.

Charles M. Schwab has been elected a trustee of Lehigh University.

O. G. Smith, associated with the Platt Iron Works Company, Dayton, O., has been made manager of the company's branch house at St. Louis, Mo.

Arthur C. Tagge, formerly identified with the Eastern Canada Portland Cement Co., Dombourg, P. Q., has become associated with the Canada Cement Co., Montreal, P. Q.

Stevenson Taylor has been elected president of the Society of Naval Architects and Marine Engineers.

Edward P. Thompson, formerly of New York, has moved his business to Washington, D. C., in order to be near the Patent Office in behalf of clients.

S. K. Thompson, formerly master mechanic with Stanley G. Flagg & Co., Philadelphia, Pa., has established an office in the Real Estate Trust Building in that city as consulting mechanical engineer.

S. Tompkins, who has been in charge of the shops and engineering department of the Miller School in Virginia, has been appointed superintendent of power

stations and chief engineer of shops and track, of the Coney Island & Brooklyn R. R., Brooklyn, N. Y.

Walter H. Trask, Jr., has been appointed district sales manager of the Denver Engineering Works Company, Salt Lake City. He was formerly assistant to sales manager in the company's main office in Denver.

F. J. Wood presented a paper on Multiple-Effect Distillation at the December 8 to 10 meeting of the American Institute of Chemical Engineers, held in Philadelphia, Pa.

PRESIDENT 1888

OF

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The professional papers contained in The Journal are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 FEBRUARY 1910 NUMBER 2

THE New York monthly meeting for February will be devoted to the dedication of a bronze memorial tablet to Dr. Robert H. Thurston, the first president of The American Society of Mechanical Engineers. All associates and former students of Dr. Thurston are earnestly invited to attend these exercises to show their esteem for him as a friend and in recognition of his brilliant career as an engineer and educator.

Addresses will be given upon Dr. Thurston as a man, and his life work, by speakers of wide reputation who knew him intimately. These addresses will touch upon his experience as an engineer of the navy during the Civil War; his work as an educator at Stevens Institute of Technology and at Cornell University; his achievements as engineer and investigator; as an author; and his long relationship with The American Society of Mechanical Engineers.

Among those who will participate are Prof. John E. Sweet, closely associated with Dr. Thurston in the organization of the Society; Col. E. A. Stevens, the prominent representative of the Stevens family, founders of Stevens Institute; President J. G. Schurman of Cornell University; and Mr. William Kent, consulting engineer. Dr. Alex. C. Humphreys, president of Stevens Instutite,will be the chairman.

The beautiful memorial which is to be unveiled is the work of Herman H. McNeil, a former student and personal friend of Dr. Thurston. It is a replica of the memorial tablet presented to Sibley College, Cornell University, by alumni and students of the university. The tablet was placed in the rooms of the Society through the gener-

osity of members and their devotion of Dr. Thurston. Contributions were received by a committee consisting of John Fritz, S. W. Baldwin, Prof. R. C. Carpenter, W. C. Kerr, E. A. Uehling, Wm. Hewitt, and Gus C. Henning, The installation of the memorial and the arrangement for the dedicatory exercises were made by a committee consisting of Dr. Alex. C. Humphreys, *Chairman*, and Messrs. Chas. Wallace Hunt, Fred J. Miller, Prof. R. C. Carpenter and J. W. Lieb, Jr.

MEETING IN BOSTON, FEBRUARY 16

There will be a meeting of engineers in Boston on February 16 conducted by the American Institute of Electrical Engineers with the coöperation of The American Society of Mechanical Engineers and the Boston Society of Civil Engineers. The meeting will be held in the auditorium of the Boston City Club, 9 Beacon St. The subject of the meeting is Industrial Power, arranged for by the Industrial Power Committee of the American Institute of Electrical Engineers. Five papers will be presented, the authors being Prof. D. C. Jackson, Mem. Am. Soc. M. E., Charles T. Main, Mem. Am. Soc. M. E., Robt. S. Hale, Mem. Am. Soc. M. E., Geo. H. Stickney and W. B. Nye.

SPRING MEETING, ATLANTIC CITY, MAY 31–JUNE 3

The Spring Meeting of The American Society of Mechanical Engineers will be held this year as usual, in addition to the London Meeting which occurs in July. Atlantic City, N. J., has been selected by the Meetings Committee and approved by the Council as the place, and the meeting will be held from May 31–June 3 inclusive. The headquarters during the meeting will be at Hotel Marlborough-Blenheim.

NEW YORK MEETING, JANUARY 11

The New York monthly meeting for January drew out a profitable discussion on lubrication. The Society was fortunate in having for its guests Dr. C. F. Mabery. Professor of Chemistry at Case School, Cleveland and Dr. P. H. Conradson, chief chemist of the Galena-Signal Oil Company, Franklin, Pa. Dr. Mabery presented a paper, published in this number of The Journal, on Lubrication and Lubricants. The paper deals largely with laboratory tests in the performance of which Dr. Mabery has been signally successful and from which

he has deduced interesting results both withoils alone, and with oil and graphite and water and graphite.

Following Dr. Mabery's address, a paper by Prof. F. H. Sibley upon Efficiency Tests of Lubricating Oils, published in The Journal for November, was read by Dr. Charles E. Lucke. The discussion was lead by Dr. Conradson, who sought to show the extent to which laboratory practice might be expected to have a bearing on the performance of lubricants in actual practice and explained certain practical considerations that must be taken into account in the lubrication of different types of machinery. Others who contributed to the discussion were William M. Davis of Boston, Henry Souther of Hartford, Conn., F. R. Low. Dr. D. S. Jacobus, C. A. Hague, and George A. Orrok of New York.

MEETING OF THE COUNCIL

A meeting of the Council was held Tuesday, January 11, in the rooms of the Society. There were present, Charles Whiting Baker, Prof. R. C. Carpenter, George M. Bond, Charles Wallace Hunt, Dr. Alex. C. Humphreys, James Hartness, Prof. F. R. Hutton, I. E. Moultrop, Col. E. D. Meier, Jesse M. Smith, F. W. Taylor and H. G. Stott, and the Secretary. In the absence of the President, Dr. Humphreys was chosen Chairman.

The minutes of the meeting of December 10 were read and approved. The Secretary announced the following appointments by the President: Executive Committee, Dr. Alex. C. Humphreys, *Chairman*, Charles Whiting Baker, *Vice-Chairman*, H. L. Gantt, Prof. F. R. Hutton, F. M. Whyte; Standing Committees: Finance, A. M. Waitt, reappointed; House, H. R. Cobleigh; Library, Alfred Noble; Meetings, Willis E. Hall, reappointed; Membership, Theodore Stebbins; Publication, Geo. M. Basford; Research, James Christie, reappointed, and in place of Dr. Charles B. Dudley, deceased, Ralph D. Mershon.

Voted: To confirm the Executive Committee as named.

The resignations of Carl S. Dow, W. A. McFarland and R. Raymond were accepted.

Voted: To approve the action of the Meetings Committee in the appointment of the following sub-committees: on Sugar Machinery, H. deB. Parsons, Thos. F. Rowland and Dr. D. S. Jacobus; on Machine Shop Practice, L. R. Pomeroy, Prof. Walter Rautenstrauch and John Parker Illsley.

Voted: That the resignations of the Committee on Power House Piping be accepted.

Voted: To accept the invitation of the National Civic Federation to this Society, to be represented at the conference in Washington, January 17–19. In accordance with a vote of the Council, the Chairman appointed the following Honorary Vice-Presidents: Jesse M. Smith, Past-President, Chas. Kirchhoff, A. W. Burchard, E. G. Spilsbury, F. M. Whyte and Wm. H. Wiley.

Voted: To accept the resignations of the Committee on Land and Building Fund, in accordance with the request of the committee.

Voted: That the Executive Committee be requested to nominate to the Council a committee of three or more to take up this work; and that such recommendation be presented at the next meeting of the Council.

In accordance with previous discussion by the Council the following amendments were formally approved:

B 23 The Finance Committee shall consist of five Members or Associates The term of office of one member of the Committee shall expire at the end of each Annual Meeting. This committee shall, under the direction of the Council, have a supervision of the financial affairs of the Society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent public accountant. The committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for presentation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate for the probable income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and, when called upon by the Council, advise upon financial questions. It shall have charge of the making of all contracts and other obligations to pay money in the Society's work and the ordering of all expenditures thereunder.

B 25 The Publication Committee shall consist of five Members or Associates. The term of office of one member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers or discussions, or parts of the same shall be printed in the Transactions of the Society. The Committee shall have the supervision of the monthly publication of the Society known as "The Journal." The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

On behalf of the Executive Committee, Charles Whiting Baker reported that the S. S. Celtic, sailing July 16, had been selected as the

vessel on which the main party attending the Joint Meeting in England would cross.

I. E. Moultrop, Chairman of the Committee on Meetings of the Society in Boston, reported regarding the joint meeting with the American Institute of Electrical Engineers and Boston Society of Civil Engineers, to be held January 21, and presented in the name of his committee an invitation to members of the Council to be present.

On motion the meeting adjourned to February 8.

JOINT MEETING WITH THE INSTITUTION OF MECHANICAL ENGINEERS, BIRMINGHAM, ENGLAND, JULY 26-29,1910

The Society has selected as the official steamer for the members and their families, *the mammoth twin-screw S. S. Celtic of the White Star Line,* which is scheduled to sail from *New York, Saturday, July 16, 1910, at 2 p.m. for Liverpool,* calling en route at Queenstown and Holyhead.

In order that we may retain our option of the *entire first-class accommodations of the Celtic,* it is necessary that all arrange to sail on this steamer.

THE OFFICIAL STEAMSHIP

The Celtic, 20,904 tons, ranks among the largest steamers in the world. Because of her exceptional steadiness and the general roominess of her staterooms and the public apartments, she is one of the most desirable of Atlantic steamers.

There is a large variety of passenger accommodations, among them several promenade and upper promenade suites, consisting of bedrooms and sitting rooms, with private bath and toilet rooms. A limited number of single staterooms, for the sole occupancy of one passenger, may be had; and there are numerous outside and inside cabins at various prices. The four promenade decks present unexcelled opportunities for rest in a steamer chair or exercise and games on deck.

The Celtic is fitted with Marconi wireless, submarine signaling apparatus and other modern safety devices.

SPECIAL RATES

From the regular tariff rates of the Celtic, the Committee has secured for our members a reduction of ten per cent, except when such a rebate would cause the price to fall below $97.50, the fixed minimum rate. For example, the rate for room 117 is $300 when occupied by two persons; but with the 10 per cent rebate the price will be $270 for two, or $135 each.

EARLY DECISION ESSENTIAL

As our option upon the Celtic's accommodations is necessarily limited because of the great pressure from the general public to leave America on this popular ship and date, all who decide to sail should communicate promptly with the *White Star Line, 9 Broadway, New York*, where all correspondence should be sent. The decision must be made before February 15.

HOTEL AND RAIL ARRANGEMENTS

The meetings will begin in Birmingham on Tuesday, July 26, and complete arrangements will be made for landing the entire party, Sunday, July 24, in Liverpool, and conveyance to and reservations in a hotel in that place. Monday will be spent in Liverpool; on Tuesday morning, the party will be conveyed by a special train to Birmingham and located in hotels in the latter city.

SIDE TRIPS

Side trips in England, on the Continent, or to any part of the world, can be arranged through the White Star Line, which will gladly reply to all inquiries. If members will immediately indicate their preferences in this matter, the White Star Line will act as a clearing house to bring together those who may have similar intentions.

THE RETURN TRIP

As the rush of return travel from Europe to America always taxes all available passenger accommodations between August 15 and September 25, members are strongly advised to secure round trip tickets now. For those who desire to return by any of the following lines, namely, White Star Line, American Line, White Star-Dominion, Atlantic Transport, Leyland and Red Star Line, the International Mercantile Marine Company (which controls them) is prepared now to make reservations *at regular rates*. The sailing dates of the various steamers will be furnished on application to the White Star Line. The bookings for European travel are the heaviest in history, and failure to reserve return passage immediately may result in serious inconvenience. Should members desire to return by other lines, the return tickets are interchangeable, but the necessity to reserve accommodations now is imperative in any case.

As the Society is the guest of the Institution of Mechanical Engineers, it will be impossible to include in the party gentlemen guests. The invitation which the Society accepted is extended only to the members and their immediate families.

On the last page will be found a list of members who have already signified their intention to attend.

Correspondence regarding the outgoing passage, reservations, side trips, etc., should properly be conducted direct with the White Star Line, 9 Broadway, New York. On the other hand the committee will be pleased to answer any communications.

Alex. C. Humphreys, *Chairman*
Charles Whiting Baker, *Vice Chairman*
H. L. Gantt
F. R. Hutton
F. M. Whyte
Executive Committee

Calvin W. Rice, *Secretary.*

ATTENDANCE AT THE JOINT MEETING

The following members, accompanied by 137 ladies, have signified their intention to attend:

The President

George Westinghouse

Past-Presidents

Prof. F. R. Hutton, *Honorary Secretary* Oberlin Smith Jesse M. Smith
Ambrose Swasey F. W. Taylor Worcester R. Warner

Vice-Presidents

W. F. M. Goss Col. E. E. Meier F. M. Whyte

Managers

H. L. Gantt James Hartness

Treasurer William H. Wiley

Chairman Membership Committee Willis E. Hall

Secretary Calvin W. Rice

E. T. Adams
Edwin H. Ahara
John G. Aldrich
C. J. Angstrom
G. Ayres
Abram T. Baldwin
C. Kemble Baldwin
William J. Baldwin
S. G. Barnes
Edward P. Bates
Charles L. Bauer
Laurence V. Benet
Wm. P. Bettendorf
Sydney Bevin
C. H. Bierbaum
F. B. Bigelow
Charles W. H. Blood
J. H. Bloomberg
Robert W. Boenig
R. P. Bolton
Wm. T. Bonner

George A. Boyden
Geo. M. Brill
Morgan Brooks
John Calder
Henry W. Carter
David A. Chapman
A. G. Christie
A. W. Colwell
Jas. V. V. Colwell
Frederick N. Connet
Geo. M. Conway
Morris Llewellyn Cooke
J. C. Cromwell
F. Daugherty
Charles Ethan Davis
F. W. Dean
D. de Lancey
James B. Dillard
Henry B. Dirks
W. F. Dixon
William T. Donnelly
Walter C. Durfee
H. Emerson
Q. N. Evans
Thomas M. Eynon
John P. Faber
A. D. Finley
H. D. Fisher
F. A. Flather
B. P. Flint
E. H. Foster
William Fox
Harry C. Francis
Lawford H. Fry
R. W. Fuller
Francis E. Galloupe
E. A. Garratt
William Gleason
F. A. Goetze
Geo. E. Hallenbeck
Chester B. Hamilton, Jr.
J. A. Herrick
Henry Hess
C. P. Higgins
M. P. Higgins
E. L. Hill
Thos. Hill
Lewis G. Howlett
Leigh A. Hunt
Dugald C. Jackson
E. L. Jahncke
Herman G. Jakobsson
E. H. Jewett
William J. Keep
L. H. Kenney
J. G. Kingsbury
Charles Kirchhoff
Frank B. Klock
G. L. Kothny
H. M. Lane
Nisbet Latta
R. K. LeBlond
Wilfred Lewis
J. H. Libbey
Wm. Lodge
Charles Longstreth
F. R. Low
Robert T. Lozier
Walter MacGregor
H. B. MacFarland
J. Macfarland
Robert A. McKee
James W. McLaughlin
Charles T. Main
A. K. Mansfield
Thos. Marrin
W. C. Marshall
R. E. Mathot
A. V. Matlack
Geo. Mesta
E. W. Mix
W. O. Moody
Robert C. Monteagle
L. H. Morgan
R. L. Morgan
James W. Nelson
Charles Z. Newell
J. G. O'Neil
Geo. A. Orrok
Henry S. Otto
Wm. F. Parish, Jr.
F. A. Park
J. C. Parker
F. A. Parkhurst
Charles H. Parson
C. D. Pettis
H. Hobart Porter
Jos. G. Prosser
Thos. C. Pulman
L. S. Randolph
E. Howard Reed
Joseph Reid
Julian Richmond
Addison A. Righter
J. M. Robinson
J. W. Roe
W. F. Rogers
Axel Sahlin
E. K. Sancton
Thomas H. Savery
C. H. Schlachter
Geo. Schuhmann
Arthur C. Scott
Alonzo B. See
E. C. Sickles
C. C. Simpson
Alton L. Smith
A. Parker Smith
Gubert S. Smith
H. F. Smith
F. H. Stillman
C. W. Stone
E. B. Stone
K. J. Sunstrom
H. H. Suplee
Frank H. Taylor
J. T. Taylor
F. W. Teele
B. L. Thompson
F. Thuman
Edw. D. Thurston, Jr.
John T. Tiplady
F. E. Town
H. P. Townsend
G. R. Tuska
Willard C. Tyler
John W. Upp
T. A. Van Der Willigen
Edward Van Winkle
P. V. Vernon
F. L. O. Wadsworth
Charles Wald
Adolph O. Wallichs
William Watson
H. H. Westinghouse
William Wilke
F. O. Willhofft
C. N. Wills
Robert York

ENGINEERS' DINNER AT BOSTON

On Friday evening, January 21, upwards of 425 engineers, representative of the engineering profession as a whole, attended the dinner at the Hotel Somerset, Boston, given by The American Society of Mechanical Engineers, the Boston Society of Civil Engineers, and the Boston branch of the American Institute of Electrical Engineers to the presidents of these societies, George Westinghouse, George B. Francis and L. B. Stillwell; to John A. Benzel, president of the American Society of Civil Engineers and other distinguished guests. While the attendance was mainly from Boston and vicinity, there was a large representation from New York and a considerable number from other cities. This was the largest and most enthusiastic meeting that the Boston engineers have held and it emphasized in an unmistakable way the cordial relations existing between the different branches of the profession and the earnest desire for coöperation. There were present eight presidents of engineering societies or institutions, besides prominent members of many others, including architectural and scientific societies closely identified with the work of engineers. The following is the list of guests and others seated at the head table at the dinner:

C. B. Edwards, chief engineer, Fore River Ship and Engine Building Co.; Arthur Warren; Asa M. Mattice, manager of works, Walworth Mfg. Co., S; Boston, Mass.; Lieut-Com. O. G. Murrfin of the North Dakota; Prof. C. F. Allen, of the Massachusetts Institute of Technology; G. A. King, president, N. E. Water Works Association; Prof. D. C. Jackson, Massachusetts Institute of Technology; E. A. Engler, president, Worcester Polytechnic Institute; Desmond Fitzgerald, member, Metropolitan Water Committee; John A, Bensel, president, American Society of Civil Engineers; Elihu Thomson; Geo. B. Francis, president, Boston Society of Civil Engineers; Prof. Ira N. Hollis, Harvard University, Chairman Boston Local Com. Am.Soc.M.E.; George Westinghouse, President, Am.Soc.M.E.; Charles Francis Adams; L. B. Stillwell, President, American Institute Electrical Engineers; Prof. Geo. F. Swain, Harvard University; Jesse M. Smith, Past-President, Am.Soc.M.E.; W. D. Wright, president, N. E. Street Railway club; Calvin W. Rice, Secretary, Am.Soc.M.E.; R. Clepston Sturgis, president, Boston Society of Architects; Chas. T. Main, Boston, Mass.; I. E. Moultrop, mechanical engineer, Edison Elec. Ill. Co. of Boston.

Following the dinner, C. B. Edwards, chief engineer of The Fore River Shipbuilding Company, gave a talk, illustrated by lantern slides, on The Main and Auxiliary Machinery of the Blatteship North Dakota. This is the new "Dreadnaught" of the U. S. Navy, turnbine driven of 20,000 tons capacity. Photographs were thrown on the screen of the ship under trial, of the machinery after installation, and many detail drawings were shown of the arrangement of the boilers, machinery and piping.

Prof. Ira N. Hollis of Harvard University, chairman of the committee on meetings of the Society in Boston, acted as toast master and referred to an inquiry made when he came to Boston 17 years ago as to why he had removed "to that remote corner of the country where the engineering efforts were but feeble compared with those of the West." He said the criticism was true geographically but that Boston was in fact a great center of engineering, examples of which he instanced. He then introduced the next speaker, Mr. George B. Francis. president of the Boston Society of Civil Engineers. the oldest engineering society in America, which has for some time had under discussion the question of an engineering building and clubhouse.

This matter, said Mr. Francis. has been considered by a committee of the Boston Society of Civil Engineers, but the available funds are not enough to enable this Society to build on its own account. The committee, therefore, considered the possibility of coöperation with other local societies and with local members of the national engineering societies, making the building a headquarters for the city. Within a radius of 15 miles of Boston about 5000 men are engaged in engineering and architecture and the committee has suggested that such a body might well combine and organize a stock company to control a property which should be a home for local engineers and embrace a clubhouse with restaurant, smoking rooms, sleeping rooms and other features. He outlined a plan for carrying out this idea, including provision for revenue. If it becomes evident, he said, that there is a real demand for some of the wealth and standing of Boston sufficient money can be raised to carry out the project.

Mr. John A. Benzel, president of the American Society of Civil Engineers, warmly advocated the plan proposed by Mr. Francis, saying that an engineers' club could be maintained in a dignified form, keeping well within the spirit of the profession, which would not only agreeably fill the needs for companionship, but would widen the horizon of the different members of the profession.

The plan was also approved by R. Clepston Sturgis of the Boston

Society of Architects and L. B. Stillwell, president of the American Institute of Electrical Engineers, who took the occasion also to speak happily of the relations existing between mechanical and electrical engineers and of the need for coöperation; and in his remarks he further paid a tribute to Mr. Westinghouse saying that for 40 years, during which there has been so tremendous a development of industries, he has stood in the very forefront. It has not been a matter of financial results, merely but of a multitude of inventions promoting the comfort of society and protecting it against dangers resulting from new engineering developments.

At the close of the meeting, Prof. D. C. Jackson made a motion, seconded by I. E. Moultrop and C. E. Clark, to the effect that a joint committee be appointed to consider the matter of rasing funds and making plans for the erection of an engineering building and clubhouse in Boston.

Following the speech of Mr. Stillwell, Hon. Charles Francis Adams said a few words of appreciation of Mr. Westinghouse, his intimate friend, after which Mr. Westinghouse was introduced and given a very hearty welcome, the audience rising. An abstract of Mr. Westinghouse's remarks is given below.

The committee having charge of this successful event consisted of Prof. Ira N. Hollis, chairman; Prof. Edward F. Miller and J. H. Libbey for the Mechanical Engineers; H. F. Bryant and F. H. Fay for the Boston Society of Civil Engineers and N. J Neall and J. F. Vaughan for the Electrical Engineers.

ADDRESS BY GEORGE WESTINGHOUSE

It seems fitting and logical that we should encourage closer and more intimate relations among all engineering societies, in order that we may benefit from the power and influence which comes from combined efforts, and by working on broad, generous lines cause individual and professional prejudice to give way to that healthful condition of mind so necessary to correct conceptions and actions.

For many years the tendencies have been toward large and powerful railway and industrial combinations and their constant skilful development. The very magnitude of these, with the evil practices so frequently disclosed, has so aroused the public that there is a fixed determination to establish an exacting governmental control of practically all forms of corporation in order that competition may be encouraged and not stifled, but seemingly without due regard to

the real objects in view, viz., the securing of the best public service in all forms; the best foods and goods for our daily needs; the greatest possible comfort to the masses; and as great freedom as possible from those restrictions which hinder rather than promote honest endeavors.

Fortunately there are indications, of which this gathering is one of many, that the great leaders in our affairs (as witness the meetings of the Governors of numerous States now being held in Washington) are alive to the importance of *the regulation of legislation* and the creation of sentiments which will bring business men to their senses.

The engineering societies, by like joint action, have it in their power to do much to better conditions. Probably there is no better way for them to do so than to show, from their knowledge and experience, that *unregulated competition and rivalry in business* have established conditions which have made our costs greater and rendered ideal conditions in industrial engineering matters most difficult of realization.

To make this clear, I need only call attention to the effects of this unregulated competition in one great industry—the electrical—which has grown up in less than twenty-five years. No user of electrical apparatus can fail to appreciate the advantage if when some repair part was needed certain standards had been followed, but it is a lamentable fact that with the single exception of uniform bases for incandescent lamps, there are now practically no standards.

To illustrate the points I have tried to bring to your attention, I quote as follows from a letter on this subject:

To illustrate the growth in the number of motor ratings required now as compared to the earlier days when sixty ratings sufficed, a summary is given of the motors manufactured by the company with which I am connected. *These figures refer to stationary motors only in sizes up to 200 h.p.* All of these motors are regularly manufactured and no special motors are included.

For direct current, 55 frames are used giving 1600 ratings.
For alternating current, 80 frames are used giving 1950 ratings.
Or a total of 135 frames are used giving 3550 ratings.

Practically anyone of these may be furnished in three types: (*a*) shaft horizontal, (*d*) shaft vertical, (*c*) with counter shaft bracket and bearings mounted on the frame. This makes a total of three times 3550, or something over 10,000 different motors available. In spite of this, there is a constant and increasing demand for special motors. In the past year approximately 10,000 estimates have been made on special motors under 200 h.p., even though the greatest effort has been made to divert all inquiries to our regular lines of motors.

Many of these special estimates were necessary because the prospective customer wanted a motor having the same characteristics as a motor offered by some other manufacturer. Our standard motor may have differed in any one of the following characteristics: *a* Horse power or speed rating; *b* Dimensions of base; *c* Overall dimensions; *d* Height from base to center of shaft; *e* Weight; *f* Method of lubrication; *g* Size of shaft; *h* Performance guarantees.

This demand for special apparatus places a heavy burden on the manufacturer. The purchaser also suffers because of increased cost and long deliveries.

Consideration of the above and a general review of the situation leads to the conclusion that the benefits that will result from standardization will more than compensate for the work and expense required in making the necessary changes.

While these particulars relate only to a part of the motors made by one large company, it must not be forgotten that there are half a hundred others manufacturing equivalent lines of motors and that each maker has his own patterns and designs, so that it is safe to say there are fifty or more thousands of needless variations in motors which have added many millions of dollars to the investment already made in installations of electrical machinery.

I have long believed, and have urged upon my associates, for the fourteen years during which the two large electrical companies have had the joint use by a license agreement of several thousands of patents relating to their business, that by coöperation in the development of apparatus, by the use of the same designs, and by the exchange of engineering and manufacturing particulars, there would be evolved the very best of all kinds of electrical machinery and details, and that the products of the two companies could be made interchangeable, not only to the advantage of purchasers and users of electrical apparatus, but also to the advantage of the companies themselves. Other views have prevailed, however, and there has existed an unregulated competition which has made the electrical industry about the poorest of all in the matter of profits.

There are many here who know of the consequences of the adoption of different gages for railways, the final result being a change involving enormous expense to those railways having the disadvantage of having adopted a standard which differed from that of their more powerful neighbors. Unless there be some action in the near future, by those who have the gift of foresight, we shall soon have a like difficult condition to meet, due to the establishing of widely different systems for the electrification of our main railways. It seems certain that a system capable of universal use should be selected in the near future so that an electric locomotive or car of one railway could operate upon all other lines.

By a combined effort of all of the engineering societies, with the financial support of all manufacturers, who would be largely benefited and could well afford to pay in pro rata amounts the expense of a well-equipped and officered bureau of standardization, it seems to me such a bureau could be established, and could work a reform of incalculable value in our present practice, thus forestalling governmental activity in this direction.

The public needs no further incitement to the regulation of such matters by the Government. What is needed is such wise cooperation on the part of the large interests involved, and such fair consideration of the public rights, as may stay further governmental action and finally render it unnecessary.

GENERAL NOTES

LUNCHEON TO CHARLES KIRCHHOFF

Mr. Charles Kirchhoff, Mem.Am.Soc.M.E., recently retired as Editor of The Iron Age after a career of marked success in technical journallism for a period of thirty years, was tendered a luncheon at the Engineers' Club, New York, on Sunday afternoon, January 16, by a large number of engineers and personal friends. Mr. Kirchhoff will take a much deserved rest which will begin with a West Indian cruise; and while the immediate purpose of the gathering was to wish him a pleasurable voyage it was in reality an outward expression of the esteem in which he is held by his many friends and a recognition of his high accomplishments in his chosen field of professional work.

The luncheon was arranged by Philip T. Dodge, president of the Engineers' Club; T. C. Martin, Chairman Executive Committee, Museum of Safety and Sanitation; E. C. Brown, past-president, American Trade Press Association; Joseph Struthers, assistant secretary, American Institute of Mining Engineers: Theodore Dwight, secretary of committee. President Dodge presided at the luncheon.

Mr. Geo. W. Cope, the present editor of the Iron Age, spoke from an association of more than a quarter of a century with Mr. Kirchhoff, paying him a personal tribute and speaking of his quick perception of the bearing of new developments in commercial or technical progress and of his ability to inspire those around him with fresh zeal and interest in their work. He presented to Mr. Kirchhoff from his former associates a French bronze statute by Picault, entitled "La Source du Pactole." A figure typifiying the engineer holding dividers and hammer is pouring from an earthern jar a stream representative of the River of Pactolus, famed for the gold carried in its sands. The bronze was given as emblematical of the effective way in which its recipient, through his profession as engineer and editor, had contributed to the material benefit of those who had come under the influence of his publication in which is recorded the product of his life's work.

Among those who spoke was John Fritz of Bethlehem, Pa., Hon. Mem.Am.Soc.M.E., who, in spite of his advancing years came to

pay his respects. He expressed his high appreciation of the part contributed by Mr. Kirchhoff to the American iron trade. In the 1840's, pig iron production in the United States was less than 300,000 tons. In a recent year 27,000,000 tons was passed and now the country is producing at the rate of 32,000,000 tons a year. As editor of The Iron Age Mr. Kirchhoff had commented on developments in the industry year after year, described the best practice, analyzed the statistics of production, kept all in the industry informed as to the work of mechanical, metallurgical and electrical engineers in this country and abroad.

Many letters of appreciation were received from friends unable to attend. Among those read were letters from Andrew Carnegie, Hon.Mem.Am.Soc.M.E.; Ambrose Swasey, Past-President Am.Soc. M.E.; Chas. Whiting Baker, Vice-President Am.Soc.M.E.; Prof. Henry M. Howe; John Hays Hammond; John W. Lieb, Jr.; Mem. Am.Soc.M.E.; Robt. W. Hunt, Past-President Am.Soc.M.E. and Hon. William H. Wiley, Treasurer Am.Soc.M.E.

The following members of The American Society of Mechanical Engineers were in attendance: Ed. A. Uehling; Henry R. Towne; J. Waldo Smith; C. H. Zehnder; Colin C. Simpson; C. M. Wales; Henry D. Hibbard; F. A. Halsey; Wm. Schwanhausser; H. R. Cobleigh; J. M. Sherrerd; Albert W. Jacobi; W. L. Saunders; E. G. Spilsbury; Jesse M. Smith; W. H. Taylor; W. W. Macon; J. E. Denton; Walter Wood; John Fritz; Calvin W. Rice; H. F. J. Porter; S. S. Webber; Alex. C. Humphreys; Theo. Stebbins; W. H. Fletcher; Col. E. D. Meier; H. H. Suplee; Dr. Richard Moldenke.

WORCESTER ECONOMICS CLUB

At the fortieth annual meeting of the Worcester Economic Club, held January 13, at Worcester, Mass., Calvin W. Rice, Secretary, made an address on the topic of the evening, The Conservation of Natural Resources. He said in part:

Natural resources are essentially national resources, hence a subject pertaining to our national welfare should enlist the interest of every citizen.

Our natural resources are of two general classes, those capable of renewal, such as forests and those which may not be replenished, such as the minerals. Obviously the intelligent use of the latter is to make them go as far as they will, improving each year as we do in our methods.

We cannot proceed much further now without realizing that in the use of these resources we must recognize that each of us is a member of society, and that, after all, the fundamental problem is that of the individual versus

society. Where the political stops and the ethical begins, unnecessarily complicates the discussion, and I will not attempt it. Suffice to say, the individual may in the long run succeed only as the community succeeds; therefore, these two phases of the question may be considered together.

Mr. Rice was followed by Dr. George F. Swain, Mem.Am.Soc.M.E. professor of civil engineering at Harvard University, who argued that community rights must prevail over individual rights on conservation or the purpose of good government would be defeated. He gave statistics to prove the advantage of centralization of power companies over small companies.

Hon. Harvey N. Shepard represented the Appalachian Mountain Club and plead for conservation of the forests, with particular reference to the White Mountain conservation. He claimed that cutting the forests tended to fill the rivers with silt.

About 275 guests were in attendance at the meeting, which followed a dinner given by the Club.

FUNERAL OF STEPHEN W. BALDWIN

Honorary Vice-Presidents, appointed to represent the Society at the funeral of Stephen W. Baldwin, were George H. Barrus, Prof. I. N. Hollis, Chas. T. Main, I. E. Moultrop, Dr. C. J. H. Woodbury.

STUDENT BRANCHES

The first regular meeting of the recently organized Student Branch of the University of Wisconsin, held January 13, was addressed by Dean Goss of Purdue University. The officers of the Association are Prof. C. C. Thomas, Mem. Am. Soc. M. E., Honorary Chairman; R. N. Trane, Chairman; E. L. Kastler, Vice-Chairman; G. A. Glick, Secretary; J. S. Langwill, Assistant Secretary; R. A. Reudenbusch, Treasurer. A copy of the constitution has been received by the Society and placed on file.

The Stanford Mechanical Engineering Society of Stanford University holds bi-weekly meetings and during the past semester there have been papers on Grounding Devices by J. B. Bubb, The Hydroelectric on the Stanilaus River by E. A. Rogers, The Mechanics of the Aëroplane by Prof. W. F. Durand. Mem. Am. Soc. M. E., and the Electric Locomotive vs. the Steam Locomotive by Prof. S. B. Charters, among others. The officers of the society are Prof. W. F. Durand, Honorary Chairman; E. A. Rogers, President; A. F. Meston, Vice-President; H. C. Warren, Secretary-Treasurer.

OTHER SOCIETIES

AMERICAN SOCIETY OF CIVIL ENGINEERS

The American Society of Civil Engineers held its 57th annual meeting, commencing January 19, 1910, in the Society House in New York. The officers elected for the ensuing year were: President, John A. Bensel, New York; Vice-Presidents, J. T. Fanning, Minneapolis, Minn., Hunter McDonald, Nashville, Tenn.; Treasurer, Joseph M. Knap, New York; Directors, Wm. E. Belknap and Horace Loomis of New York, Geo. A. Kimball, Boston, Percival Roberts, Jr., Mem. Am.Soc.M.E., Philadelphia, Chas. F. Loweth, Chicago, Arthur D. Foote, Grass Valley, Cal.

In addition to the business sessions, the program included excursions to the new terminal station of the Pennsylvania Railroad of New York City and to the Ashokan Reservoir of the Board of Water Supply, New York. On Thursday evening, Walter McCulloh, consulting engineer of the State Water Supply Commission of New York, gave a lecture on the Conservation of the Water Resources of New York State.

AMERICAN INSTITUTE OF MINING ENGINEERS

The spring meeting of the American Institute of Mining Engineers will be held at Pittsburg, Pa., Tuesday March 1 to Saturday March 5 with headquarters at the Hotel Shanley. This will be largely a metallurgical meeting and members of The American Society of Mechanical Engineers will be welcomed to its sessions. Invitation cards may be had upon application to the Secretary of the Institute, at 29 West 39th Street, New York.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

At the meeting of the American Institute of Electrical Engineers on January 14, Prof. W. S. Franklin and Stanley S. Seyfert of Lehigh University presented a paper on The Space Economy of the Single-Phase Series Motor. The annual dinner of the Institute will be

held at the Hotel Astor on Thursday evening, February 24, with Prof. Elihu Thomson, who has been awarded the first Edison Medal, as the guest of honor.

At the December meeting of the Board of Directors of the Institute 74 Associate members were elected and 66 Students enrolled, while in January 88 Associates and 67 Students were received. The 1910 Year Book has just been issued.

INTERNATIONAL ASSOCIATION OF REFRIGERATION

At a meeting of the Council of the International Association of Refrigeration, held in Paris, December 3, 1909, Gardner T. Voorhees, Mem.Am.Soc.M.E., was elected a member of the Council, in place of Thos. S. McPheeter, deceased, to serve until the meeting of the Second International Congress at Vienna.

It was announced that the Vienna Congress would be held in the University Buildings from October 6–11, 1910, enabling the visitors to be present at the hunting and sporting exhibitions.

WESTERN SOCIETY OF ENGINEERS

The annual meeting and dinner of the Western Society of Engineers was held January 12, 1910 at the University of Chicago. Bernard E. Sunny, president of the Chicago Telephone Company, made the principal address of the evening, on The Engineering of Chicago. Mr. Sunny outlined the various broad plans for the development of public utilities and municipal improvements in Chicago proposed during the last two or three years, embracing sewers and a high-pressure water system for the central business district, the steam railway terminals of Chicago, plans of the Board of Local Improvements for replacing pavements in the business district, deep waterways, harbors and subways. The "Chicago Plan" prepared by the Commercial Club of Chicago, was also given extended consideration, and Mr. Sunny suggested that a board of engineers be created to compile the necessary engineering data relative to such a plan.

NEW ENGLAND WATER WORKS ASSOCIATION

One hundred and fifty members and guests attended the annual meeting of the New England Water Works Association held at the Hotel Brunswick, Boston, Mass., on January 12, 1910. Papers were presented on Governmental Policy in Relation to Water Power

by Marshall O. Leighton, discussed by Dugald C. Jackson, Mem. Am.Soc.M.E., Geo. F. Swain, Mem.Am.Soc.M.E., and Dwight Porter; and on the Maidstone Typhoid Epidemic, by Wm. T. Mason, discussed by Robert S. Weston. Geo. A. King was elected president and Willard Kent, Secretary.

NECROLOGY

HORACE SEE

Horace See, President of the Society in 1888, died in New York City on December 14, 1909.

He was born in Philadelphia, and after the usual classical and mathematical education of the private school entered the shops of I. P. Morris. Thence passing to Neafie & Levy, and the National Armor and Shipbuilding Co., at Camden, and Geo. W. Snyder of Pottsville, he entered on his best known life-work with William Cramp & Sons.

He rose here to be designer and superintending engineer (in 1879), designing vessels and machinery of greatly improved construction and performance, introducing improved methods of work and standards in that great establishment, and giving to the United States a shipbuilding plant of capacity and quality to compare favorably with the products of the Clyde and Newcastle. It was under his leadership that the United States Navy contracts for the first vessels of what was then called the "New Navy of the United States" were taken, and the big ships of the American Line at that day bore his impress. It was at the zenith of this busy period, when he was confessedly the leader in his field, that the presidency of the Society was placed in his hands. He presided at the Nashville and Scranton meetings of 1888.

The following year it became apparent that avenues of professional advancement would not open further for him in Philadelphia, so that he came to New York with the honors thick upon him won from his busy years. He became at once consulting engineer for the Newport News Steamship and Dry Dock Company, and was the host of the Society on his visit to that plant at the Richmond meeting of 1890. He was superintending engineer for the Southern Pacific Company, and the Pacific Mail Steam Ship Co., superintendent for the Cromwell Steam Ship Co., and in his private practice as a marine engineer and naval architect he designed and prepared specifications for many yachts and commercial vessels. Some of his improvements in hull and machinery are in international use.

Mr. See was adjutant of the Twentieth Regiment of the National Guard of Pennsylvania during the riots of 1877, and later Captain of the First Pennsylvania Regiment. Besides various business and social connections, he was a member of the American Society of Naval Architects and Marine Engineers, of the Institute of Naval Architects of Great Britain, as well as of the Northeast Coast Institute of Engineers and Shipbuilders, and the American Geographical Society; associate member of the American Society of Naval Engineers and the United States Naval Institute; and fellow of the American Association for the Advancement of Science.

He contributed a paper on the method he introduced for producing true crank shafts for multiple-cylinder engines[1] his presidential address was a discussion of manual training and methods of instruction for technical work.

STEPHEN WARNER BALDWIN

In the death of Mr. Baldwin there has passed away another of the notable figures in the engineering history of the United States. He belonged to the era of practical training which brought forward so many gifted men in the nineteenth century, and to take part in the active developments following the Civil War.

He was born in Baldwinsville, N. Y., February 4, 1833, and received his early education in Homer in that state. He entered the Lawrence Machine Shops at Lawrence, Mass., as an apprentice under the late John C. Hoadley, dividing his three years between the machine shop, forge, boiler shop, and drawing-room. The admiration he felt for Mr. Hoadley lasted all through his life, and on the death of his old chief, both out of affection and out of sentiment, Mr. Baldwin was active in buying from the Hoadley estate a large amount of expert apparatus, which was made a gift to the Society.[2] Mr. Baldwin worked with Mr. Hoadley on his single-valve automatic engines and on his portable or farm engines. From this experience he always had a strong interest in the development of agricultural machinery for the West.

He later became manager of the Clipper Mowing Machine Works at Yonkers, N. Y., and was associated with the Johnson Iron Works at Spuyten Duyvil, N. Y., improving the machines of both of these companies with his own inventions. He soon became one of the promi-

[1] Transactions, Vol. 7, p. 521.

[2] Transactions Vol. 8. p. 349. Some of this apparatus remains in the possession of the Society as museúm specimens. Other units have been sold and loaned where thay could be made useful.

nent mechanical engineers in the field of manufacture of steel and was president of the Spaulding & Jennings Co. But he will be principally remembered as the New York representative and agent for so many busy and successful years for the Pennsylvania Steel Company and the Maryland Steel Company. He remained with these companies until 1904 when he was retired, but was an honored adviser until his death.

Mr. Baldwin was intensely interested in the problems of education for the young engineer. When Milton P. Higgins proposed his scheme of half-time schools in which lads were to work at books for half the day and in the shop atmosphere for the other half, Mr. Baldwin's interest was not alone because the projector had been a fellow apprentice at Lawrence, but because he believed the idea to be sound. The Artisan School of Syracuse, in which his friend Prof. John E. Sweet is so important a factor, was also near his heart.

Mr. Baldwin was early brought into active relations with the Society. He served on early nominating committees, was made Manager for the term 1887–1890 and Vice-President for 1890–1892, serving five continuous and important years on the Council. But his greatest service was as Chairman of the finance committee with control over the budget of each year. He was successively re-appointed fourteen times, and his service ceased only with the changes in constitution and by-laws in 1904. He was a member of the Council's Committee which bravely faced the problem of the purchase of No. 12 West 31st Street, in 1890, when the Society had no capital to invest in such a great undertaking, except the earnest purpose of those members whom the secretary of that date had stimulated to the point of venturesomeness. The bonds issued as a part of the financial scheme were all redeemed and the second mortgage paid off within the period of Mr. Baldwin's activity. He worked very hard for two winters over a plan to develop meetings of the junior members of the Society for their common advantage. The monthly meetings of the Society in different cities are an heritage from those efforts.

Mr. Baldwin was a sound and straight thinker, a man of great power of application, an analytical reasoner, a diligent and painstaking worker. Tall and commanding of figure, he had the grace, refinement and broad culture of the scholar. His inventive mind was always at work, adding labor-saving devices as well as improvements to whatever interested him, and his pleasing personality intensified the impression of good fellowship by which he put every one at ease. He inspired such confidence in his integrity that he was constantly

sought as an adviser. He was at one time a member of the American Society of Civil Engineers and of the American Institute of Mining Engineers. He was very active also with his friend J. F. Holloway in the buildimg up of the Engineers Club, and was one of its four honorary members.

Mr. Baldwin died at the home of his daughter on January 5, 1910, after several years of physical weakness, although of clear mental capacity, and a personality active in the days of the up-building of the Society has gone to his reward.

CHARLES B. DUDLEY

Dr. Charles B. Dudley died at his home in Altoona,Pa., on December 21, 1909. He was born July 14, 1842, at Oxford, Chenango Co., New York, where he received his early education. In 1862 he enlisted as a private soldier in the 114th New York Volunteers and fought in seven battles, finally receiving a severe wound at the battle of Opequan Creek in 1864. Returning from the war in 1865, he prepared at the Oxford Academy and Collegiate Institute to enter Yale, from which he received the degree of A. B. in 1871; and in 1874, the degree of Ph.D. His graduation thesis, On Lithium and a Glass made with Lithium was published in full abstract in the Proceedings of the American Association for the Advancement of Science.

The following year he became assistant to Dr. George F. Barker, Professor of Physics at the University of Pennsylvania, and during this time published in the Franklin Institute Journal some translations of German technical papers. After a month spent as teacher of sciences at Riverview Military Academy, Poughkeepsie, N. Y., in November 1875 he went to Altoona to take up his life work as chemist of the Pennsylvania Railroad.

When Dr. Dudley entered upon his new task, no railroad had a chemist as a regular employee, although many had occasional chemical work done, and the whole subject of the relation between scientific knowledge and its practical use by railroads was in a very chaotic state. It would not be possible to enumerate the special investigations and studies leading to modifications of practices in daily use on railroads which have been considered since that time by the experimental department at Altoona, for the chemical part of which Dr. Dudley was responsible. That which attracted the most widespread attention, perhaps, was the study of steel rails, made in the early eighties, which gave the steel-maker as never before a view of

his product from the standpoint of the consumer and forced upon him a study of it not only for immediate output but also with an eye to the demands which service would make upon it.

Another very important line of work has been the making of specifications, perhaps the most exacting and time-consuming undertaken. Investigations have been made, furthermore, into the questions of ventilation, car lighting, steam heating of cars, disinfectants, cast iron for car wheels and other important uses, paints, long-continued tests on bearing metals, analyses of coals, water supplied both for boiler use and drinking, and explosives.

Dr. Dudley has been abroad on three important commissions: in 1886 to study oil burning on locomotives in Russia, in 1900 as a delegate to the International Railway Congress in France, and in 1909 as a delegate to the Convention of the International Society for Testing Materials in Denmark. He had been vice-president of the American Institute of Mining Engineers, and twice president of the American Chemical Society. At the time of his death he was president of the International Society for Testing Materials, as well as of the Bureau of Explosives of the American Railway Association. He was a member of the English, French and German Chemical Societies; of the Iron and Steel Institute of Great Britain; of the Verein deutscher Eisenhüttenleute; the American Society of Civil Engineers; the American Institute of Electrical Engineers; and social clubs in Philadelphia, Washington and New York. He was also much interested in the Altoona Mechanists' Library.

Dr. Dudley was a member of the Research Committee of the Society.

WILLIAM METCALF

William Metcalf was born at Pittsburg, Pa., September 3, 1838 and educated there and at the Rensselaer Polytechnic Institute, from which he was graduated in 1858. Immediately after graduation he went into the employ of the Ft. Pitt Foundry, as draftsman and afterwards as superintendent, and later joint proprietor. One of his chief duties as superintendent was the casting of mortars, shells and guns for the United States Government during the Civil War, at a time when the largest cast-iron guns ever made were being cast at this foundry.

Soon after the close of the war, Mr. Metcalf bought an interest in the firm of Miller, Barr & Parkin, later Miller, Metcalf & Parkin, and after incorporation in 1889 known as the Crescent Steel Company.

This company engaged in the manufacture of fine steel. In 1895, Mr. Metcalf retired from the Crescent Steel Company and in 1897 organized the Braeburn Steel Company, of which he was principal stockholder and president at the time of his death, December 5, 1909. His book, Steel, a Manual for Steel Users, is regarded as an authority.

Mr. Metcalf was a member and one-time president of the American Society of Civil Engineers and the American Institute of Mining Engineers, and first president of the Engineers' Society of Western Pennsylvania. He had served as vice-president of the American Iron and Steel Association, and was a member of the Institution of Civil Engineers of Great Britain. In addition he was a member of the Duquesne Club of Pittsburg, and the Century Association and Engineers' Club of New York, and was actively engaged in hospital and charity work. He was appointed by the United States Government one of seven appraisers for the condemnation of the property and franchise of the Monongahela Navigation Company, in March 1897.

Mr. Metcalf entered the Society in 1880 and was its vice-president from 1882 to 1884.

PERSONALS

Geo. M. Brill and Horace C. Gardner have formed a partnership under the name of Brill & Gardner, continuing the engineering and architectural practice heretofore conducted by Mr. Brill. Mr. Gardner was formerly manager of the construction and mechanical departments of Swift & Co. The offices will be in the Marquette Building, Chicago, Ill.

J. Ansel Brooks delivered an illustrated lecture on Aerial Navigation at the January 11 meeting of the Brown University Engineering Society, formerly known as the Brown University Society of Civil Engineers.

Prof R. C. Carpenter and E. H. Faile announce a partnership for the practice of engineering, with office at 68 William Street, New York. This change will not prevent the continuation of his duties at Cornell University by Professor Carpenter, who will take up his work of instruction again at the expiration of his present year's leave of absence from the University. Mr. Faile was formerly associated with the City Investing Company, New York.

A. C. Dinkey has been made a member of the board of trustees of the Carnegie Library, Pittsburg, Pa.

George W. Dunham has severed his connections with the Hudson Motor Car Company, and is now occupying the position of vice-president and consulting engineer, with the Chalmers-Detroit Motor Company, Detroit, Mich.

Wm. Wood Estes, of Providence, R. I., has taken a position with the chief engineer of the Rhode Island Co.

Edwin. J. Haddock has given up his office in Columbus, O., to accept a position with the Tennessee Coal, Iron and Railroad Company, as mechanical and structural engineer in the coal mining department.

Sir R. A. Hadfield has been elected a vice-president of the Faraday Society of London, for the next session, 1909-1910.

John T. Horton, formerly manager of the Dobbie Foundry and Machine Company, New York, has opened an engineering office at 95 Liberty Street, New York, specializing on machinery and appliances for hoisting and handling material and contractor's equipment.

Frederick H. Keyes, formerly general manager of the Robb-Mumford Boiler Company, has associated himself with Messrs. Timothy W. Sprague, Henry D. Jackson and others, to conduct a general consulting engineering practice in New York.

Walter Laidlaw, originally identified with the Laidlaw-Dunn-Gordon Co., Cincinnati, O., and general manager of the Snow Steam Pump Works, Buffalo, N. Y., is hereafter to be located at the New York office of the International Steam Pump Co.

J. W. Lieb, Jr., was elected vice-president of the National Society for the Promotion of Industrial Education at its annual convention at Milwaukee, December 2–4, 1909.

Walter M. McFarland, acting vice-president of the Westinghouse Electric and Manufacturing Company, East Pittsburg, Pa., has resigned to engage in other business.

W. K. Millholland, until recently secretary of the international Machine Tool Company, Indianapolis, Ind., has formed the W. K. Millholland Machine Company, Indianapolis, of which he is president.

Charles E. Rogers has become connected with the Johannesburg, South Africa, office of Fraser & Chalmers, Ltd. Until recently he was associated with the Melbourne, Australia, office.

L. H. Thullen, who has conducted a consulting practice in electrical engineering in New York, has recently accepted the position of chief engineer with the Triumph Electric Company, Cincinnati, O.

W. R. Warner presented a paper on Egypt and the Pyramids at the January 11 meeting of the Cleveland Engineering Society.

Earl Wheeler has resigned his position as director of the department of electrical and mechanical engineering, Engineer School, United States Army, to become electrical and mechanical engineer of the Electric Speedometer and Dynamometer Manufacturing Company, 1317-1319 New York Ave., Washington, D. C

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OF
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The professional papers contained in The Journal are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 MARCH 1910 NUMBER 3

THE New York monthly meeting for March will be held in the Engineering Societies Building, Tuesday evening, March 8, the American Institute of Electrical Engineers participating. The paper will be by H. G. Stott, Mem.Am.Soc.M.E., Superintendent of Motive Power, Interborough Rapid Transit Company, New York, and J. S. Piggott, on Tests of a 15,000-kw. Steam-Engine-Turbine Unit.

The paper relates to the installation of low-pressure turbines at the 59th Street station of the Interborough Rapid Transit Company, New York, and presents a discussion of the most important development in steam engineering since the introduction of the steam turbine. The station is equipped with engines of the Manhattan type, which are double engines having 42 in. horizontal high-pressure cylinders and 86 in. vertical low pressure cylinders with a 5000-kw. generator. The generator is capable of carrying a load of 8000 kw. continuously but the best economy of the engine is obtained at about 5000 kw. and a low-pressure turbine was added to operate on the exhaust steam from the engine, with a view to increasing the capacity of the unit and at the same time improving the efficiency. Two turbines have been installed and the third is in process of installation. By the addition of the turbine the engine can be run to the full capacity of the generator to which is added the current from the turbo-generator, making a total output of 15,000 kw. The paper is published in this issue and gives complete details of the results of tests. An extended discussion is expected by prominent engineers connected with central stations.

BOSTON MEETING, MARCH 11

The next monthly meeting in Boston will be held Friday evening, March 11 in the auditorium of the Edison Electric Illuminating Company. The Boston Section of the American Institute of Electrical Engineers will cooperate in the meeting, and it is expected also that the Boston Society of Civil Engineers will join with these organizations. The paper will be by M. W. Alexander, member Am.Soc.M.E., who has been so long identified with the educational work and training of apprentices and employees at the works of the General Electric Company, West Lynn, Mass. The subject of the paper is The Training of Men, A Necessary Part of the Modern Factory System. This paper was published in the January number of the Journal.

MEETING IN BOSTON, FEBRUARY 16, 1910

A meeting of the American Institute of Electrical Engineers, The American Society of Mechanical Engineers coöperating, was held in the auditorium of the City Club of Boston, February 16. At an informal dinner held at the club preceding the meeting, 250 members and guests were present, while about 500 attended the professional session. The meeting was called to order at 8 o'clock by Prof. Dugald C. Jackson, Mem.Am.Soc.M.E., chairman of the Boston section of the American Institute of Electrical Engineers. David B. Rushmore, Mem.Am.Soc.M.E., chairman of the Industrial Power Committee of the Institute was called to the chair and presided during the presentation of the following papers: The Applicability of Electrical Power to Industrial Establishments, by Dugald C. Jackson, Mem. Am.Soc.M E ; Central Stations vs. Isolated Plants for Textile Mills, by Charles T. Main, Mem.Am.Soc.M.E.; The Supply of Electrical Power for Industrial Establishments from Central Stations, by R. S. Hale, Mem.Am.Soc.M.E.; Illumination for Industrial Plants, by G. H. Stickney; The Requirements for an Induction Motor from the User's point of View, by Walter S. Nye. The discussion was principally upon Mr. Main's introductory paper, Central Stations vs. Isolated Plants for Textile Mills. The meeting was successful in every way, and an indication of the wisdom of engineers in all branches getting together to discuss topics of general interest.

YEAR BOOK FOR 1910

The Year Book of the Society for the year 1910 is now being distributed to the membership. It is issued in new form, designed to

embody the advantages both of the Year Book, as previously issued, and of the Pocket List, which has formerly been published in August of each year.

The Year Book originally had two lists of members, one alphabetical, containing particulars regarding the business, the membership in the Society, and the business and home addresses of each member; and a geographical list, containing only the names of the members. The Pocket List was arranged with the geographical list containing the details regarding the members and an alphabetical list giving only the names, without other information.

In its new form, the Year Book contains a complete alphabetical list as before, and in addition, a geographical list with sufficient information regarding the business title and address of each member to make the list useful to one traveling or to those who desire to correspond with members in any particular city or connected with a particular firm.

The book is issued in a size that is convenient either for the desk or to carry in traveling and is bound in substantial cloth-covered board covers.

STUDENT BRANCHES

UNIVERSITY OF KANSAS

At the first annual meeting on December 9, 1909, papers were presented by S. M. Manley, Mem.Am.Soc.M.E., on A Ten Hour Log of a Boiler Plant; John D. Garver, Student, 1909, on The South American Machinery Market; Louis Bendit, Mem.Am.Soc.M.E., on Economical Power Development; P. F. Walker, Mem.Am.Soc. M.E., on Testing Lubricating Oils; and Paul M. Chamberlain, Mem.Am.Soc.M.E., on Increased Efficiency in the Boiler Room. An opportunity was given to visit plants and buildings in connection with the meeting and the business sessions were followed by a dinner to members and guests. Mr. Chamberlain in his paper, now on file in the headquarters of the Society in New York, discussed at length important features entering into the design and operation of boilers and furnaces, present costs of apparatus and of power plants complete, and the cost to operate coal and ash handling machinery. The organization holds weekly meetings at which technical papers and magazine reports are presented by students, as well as by occasional invited speakers.

PENNSYLVANIA STATE COLLEGE

The first regular meeting of the Pennsylvania State College Student Section as held on January 14, with Aeronautics as the topic for discussion. Prof. Arthur J. Wood, Mem.Am.Soc.M.E., exhibited a small model biplane which he had constructed. At the February meeting Power Plant Accessories were considered.

BROOKLYN POLYTECHNIC INSTITUTE

The student branch of the Brooklyn Polytechnic Institute is preparing to publish an annual, to contain papers by speakers, professors and students. At the January meeting Leon B. Lent, Mem. Am.Soc.M.E., delivered an illustrated lecture on Modern Gas Engines.

SPRING MEETING, ATLANTIC CITY

The Spring Meeting of the Society will be held at Atlantic City, May 31 to June 3 inclusive, with headquarters at the Marlborough-Blenheim. There will be the usual professional sessions, announcement of which will be made later, and on Wednesday evening, June 1, honorary membership will be conferred upon Rear-Admiral Geo. Wallace Melville, U. S. Navy, Retired, and Past-President of the Society.

RAILROAD TRANSPORTATION NOTICE

Arrangements for hotel, transportation and Pullman car accommo dations should be made personally.

Special concessions have been secured for members and guests attending the Spring Meeting in Atlantic City, May 31 to June 3, 1910.

The special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below. Read item *g*.

a Buy your ticket at full fare for the going journey, between May 27 and June 2 inclusive. At the same time request a certificate, *not a receipt*. This ticket and certificate should be secured at least half an hour before the departure of the train.

b Certificates are not kept at all stations. Ask your station agent whether he has certificates and through tickets. If not, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point, and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to S. Edgar Whitaker, office manager at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after June 3.

d An agent of the Trunk Line Association will validate certificates, June 1, 2, 3. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to June 8, on the same route over which the purchaser came, at three-fifths the rate.

g Members and guests from New York City should buy the regular round trip tickets at $5 and show the return portion to Mr. Whitaker or the validating agent.

The special rate is granted only for the following.:

The Trunk Line Association:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

The Central Passenger Association:

The portion of Illinois south of a line from Chicago through Peoria to Keokuk and east of the Mississippi River, the States of Indiana, and Ohio, the portion of Pennsylvania and New York north and west of the Ohio River, Salamanca and Buffalo, and that portion of Michigan between Lakes Michigan and Huron.

The New England Passenger Association except via N. Y. O. & W. R. R. and Eastern Steamship Co. The Rutland R. R. participates in fares reading via its road:

Maine, New Hampshire, Vermont, Massachusettes, Rhode Island and Connecticut.

The Western Passenger Association offer revised one-way fares, on the basis of two cents per mile, to Chicago, Peoria and St. Louis; these three places are points in the Central Passenger Association, and from these points purchase round trip tickets, in the manner outlined in the preceding paragraphs:

North Dakota, South Dakota, Nebraska, Kansas, Colorado, east of a north and south line through Denver, Iowa, Minnesota, Wisconsin, Missouri, north of a line through Kansas, Jefferson City and St. Louis; Illinois, north of a line from Chicago through Peoria to Keokuk.

JOINT MEETING WITH THE INSTITUTION OF MECHANICAL ENGINEERS

BIRMINGHAM AND LONDON, ENGLAND, JULY 26-29, 1910

In response to the invitation of The Institution of Mechanical Engineers to the Joint Meeting of the Institution of Mechanical Engineers and The American Society of Mechanical Engineers to be held in Birmingham and London in July, 117 members and ladies have already engaged passage on the official steamship Celtic, while 205 members, with 155 ladies, have signified their intention of going.

The following tentative program has been outlined by a Committee of the Council of The Institution of Mechanical Engineers.

Tuesday, July 26, 1910. Birmingham.

Morning, 10 a.m. Reception in Birmingham. Professional Session.
Afternoon. Visits to Works, and to Stratford, etc.
Evening. Garden Féte.

Wednesday, July 27, 1910. Birmingham.

Morning. Professional Session.
Afternoon. Visits to Works, etc.
Evening. Reception in Council House, by invitation of the Right Hon. the Lord Mayor of Birmingham.

Thursday, July 28, 1910.

Whole-day Excursions; arriving in London in time to reach hotels and attend late Conversazione at the Institution.

Friday, July 29, 1910. London.

Morning. Professional Session at the Institution of Civil Engineers.
Afternoon. Visits and social functions.
Evening. The Institution Dinner. Including Ladies.

Saturday, July 30, 1910, London.

Whole-day excursions, Windsor and Henley (by invitation).

Evening. Reception at the Japanese-British Exhibition. By invitation.

The Committee are of opinion that members of the American Society will prefer to make their own arrangements for the time between the arrival in Liverpool and the opening of the Meeting in Birmingham at 10 o'clock on Tuesday, as Chester and other places of interest are in the immediate vicinity.

With regard to hotels in Liverpool, the Adelphi, Exchange and Great Western are considered the three principal hotels. Southport ($18\frac{1}{2}$ miles away) is now almost a suburb of Liverpool on account of the frequent service of electric trains, and is a pleasant seaside resort. A list of hotels in Birmingham and neighborhood will be published later.

INVITATION FROM INSTITUTION OF CIVIL ENGINEERS

The following letter of invitation has been received from the Institution of Civil Engineers and acknowledged with appreciation by the Council of The American Society of Mechanical Engineers:

THE INSTITUTION OF CIVIL ENGINEERS
GREAT GEORGE ST., WESTMINSTER, S. W.
11 January, 1910.

Calvin W. Rice, Esq., Secretary,
The American Society of Mechanical Engineers,
29 West Thirty-ninth Street, New York

My Dear Sir,

Hearing from the Institution of Mechanical Engineers that a joint meeting between that Institution and The American Society of Mechanical Engineers is to be held here in July next, the Council of this Institution, at a meeting held today, desired me to request you to be so good as to convey to the members of The American Society of Mechanical Engineers a very cordial invitation to them to avail themselves, during the period of the meeting, of the accommodation which the rooms of this Institution can afford.

I am
Yours faithfully,
J. H. T. TUDSBERY,
Secretary.

REPORTS

MEETING OF THE COUNCIL

At a meeting of the Council, held on Tuesday, February 8, 1910, in the rooms of the Society, there were present Messrs. Abbott, Baker, Bancroft, Bond, Carpenter, Gantt, Hartness, Humphreys, Hutton, Meier, Moultrop, Reist, Smith, Stott, Waitt and the Secretary. Regrets were received from the President and from W. F. M. Goss, Vice-President. On motion Dr. Humphreys acted as Chairman.

The minutes of the meeting of January 11 were read and approved. The Secretary reported the death of Charles Batchelor.

The resignations of R. Carter Beverley, J. S. Avery, Jr., and F. C. Slade were accepted, and announcement of the lapsing of the membership of the following was made by the Secretary: E. E. Barnard, F. E. Bradenbaugh, Jas. Breen, J. M. Briggs, E. D. Clarage, J. C. Dodwell, W. F. Donovan, L. H. Gardner, J. N. Gregory, G. O. Hodge, Nathaniel Lombard, C. F. Meissner, E. E. Miller, Jas. Naughton, H. E. Newell, H. W. Pudan, W. B. Reed, F. A. Schroeder, E. O. Spillman, G. W. Steward, F. P. Thorp, A. A. Thresher, and A. J. Weichardt.

The Executive Committee reported that 49 members had reserved passage on the Celtic, the official steamship for the Joint Meeting in England.

Voted: That the Meetings Committee, in providing for the professional sessions, including papers and discussions, for the Joint Meeting, will entirely fulfill its duty.

The report of F. M. Whyte, Honorary Vice-President, to the National Civic Federation, was received and placed in file.

Voted: That the Secretary be directed to reply to the request for coöperation with the editors of the proposed American Year Book that the Society will be pleased to lend assistance by giving information but can take no official part directly or indirectly in the publication.

Voted: That the invitation of the Institution of Civil Engineers be accepted with thanks, whereby the courtesies of the rooms of

the Institution are extended to the members of The American Society of Mechanical Engineers attending the joint meeting with the Institution of Mechanical Engineers in July 1910.

The election of Rear-Admiral George W. Melville to Honorary Membership by unanimous ballot, was announced.

Resolved: That the checks from the Treasurer be made payable to the order of the cashier and his bond be equal to the maximum amount of funds subject to his control.

Mr. Waitt, Chairman of the Finance Committee, presented a further report regarding the proposed amendments to B18 and C18 that had been referred to the Finance Committee for recommendation and report to the Council.

Voted: That the proposed amendments with the report of the Finance Committee be referred to the Committee on Constitution and By-Laws for report to the Council.

Voted: To approve the application for a student branch at the University of Maine, Orono, Me.

Voted: To approve the modifications of the standards, as approved by the Executive Committee and referred to them under the provisions of the By-Laws, January 21, 1910: general ledger, members card ledger, collection of dues, funds of the society, instructions on savings accounts, instructions on finance report, general information for office.

Notice was also given of proposed amendments to standards on instructions for paying bills, instructions on membership, committee work, election of members, classification of accounts, style sheet, cashier's funds, purchasing, etc.

On motion the meeting adjourned to March 8, 1910.

ABSTRACT OF REPORT OF LIBRARIAN

To the Library Committee:

Books and Pamphlets in the Library during the period July 1, 1908, to December 31, 1909, are listed in the accompanying table. During this time 1531 volumes, chiefly periodicals, have been bound at a cost of $1339.92.

BOOKS AND PAMPHLETS IN THE LIBRARY

	Accessions		Total in Library	
	Books	Pamphlets	Books	Pamphlets
A. S. M. E.	562	67	8,607	1406
A. I. M. E.	1176	2638	18,119	4542
A. I. E. E.	1573	1152	13,936	1248
United Engineering Societies	37	23	37	23
Total	3348	3880	40,699	7219

PERIODICALS

On July 1, 1908, there were 690 current periodicals in the joint libraries, 259 being duplicates, and on December 31, 1909, the number of periodicals regularly received, exclusive of duplicates, was 557.

Seventy-four of the duplicate sets were sold during 1909 at a net profit of $150.21. In addition $111.16 was received from the sale of various books and odd periodicals.

Researches and transcriptions have been made during the period to the amount of $31.

CIRCULATION OF BOOKS

Call cards were placed in commision on March 1, 1909 and the following table indicates the character of the books and periodicals consulted. This tabulation, however, represents only about a third of the circulation, as many people go directly to the shelves and do not ask for information.

Architecture	6	Hydraulic Engineering	17
Bibliography	2	Mathematics	16
Biography	6	Marine Engineering	10
Chemical Technology	25	Mechanical Engineering	102
Chemistry	14	Metallurgy	65
Civil Engineering	19	Mining Engineering	52
Description and Travel	6	Miscellaneous	7
Electrical Engineering	75	Periodicals	111
Electricity	33	Physics	21
Engineering	31	Railroad Engineering	16
General Geology	83	Railroad Engineering, Electric	17
		Total	734

The attendance in the Library is shown in the following table:

LIBRARY ATTENDANCE

JANUARY 1908—DECEMBER 1909

1908	Day	Night	1909	Day	Night
January	541	148	January	485	250
February	439	241	February	533	254
March	417	203	March	536	280
April	403	210	April	529	217
May	400	196	May	462	221
June	419	136	June	484	196
July	441	....	July	472	....
August	362	....	August	472	....
September	392	125	September	434	220
October	381	180	October	471	238
November	435	200	November	479	223
December	520	441	December	545	301
Total	5151	2080	Total	5901	2402

Total for 1908: 7231. Total for 1909: 8303.

During 1909 the general reference section of the Engineering libraries has been strengthened by the addition of the following reference books:

Bartholomew's New Atlas of the World's Commerce.
Bouvier's Law Dictionary, 2 vols.
Calisch's Dictionary of the Dutch Language, 2 vols.
Cyclopedia of Building Trades, 6 vols.
Dietrich's Bibliographie der deutschen Zeitschriften Literatur.
Flügel's Universal German-English Dictionary, 3 vols.
Larousse's Dictionnaire Française, 22 vols.
Meyer's Grosses Konversation Lexikon, 20 vols.
Mullhouse's Italian Dictionary, 2 vols.
Rand & NcNally Business Atlas.
Michaelis's Portuguese-English Dictionary, 2 vols.
Webster's New International Dictionary.
Qui Etes-Vous?
New York Business Directory.
Wer Ist's?

This department is also equipped with:

The Annual Library Index, 1908 (popular engineering material arranged in classes).

The Engineering Index, which is arranged separately in binders by the following classes: civil engineering; electrical engineering; industrial economy; marine and naval engineering; mining and metallurgy; mechanical engineering; railway engineering; street and electric railways.

Engineering Digest.

Le Mois Scientifique et Industriel.

Technical Index.
Reader's Guide.
Poole's Index.
Technical Press Index.
Repertorium der technischen Journal Literatur (superseded in 1908 by Technische Auskunst, published by Bibliographical Institute of Berlin.
Stone & Webster's Current Literature.

There is also a card index of the literature in the engineering periodicals received by the Engineering Societies library. Many of these entries duplicate those of the engineering index, but the library data are available several weeks earlier than the monthly printed index.

The two front alcoves have been arranged to be of practical use to the general public. They possess more advantages than are usually permitted in any of the large public libraries.

To the present equipment will be added very soon:

a Indexes of periodicals and transactions of societies.
b All current foreign periodicals.
c Official patent publications of all countries.
d Lists of periodicals in other libraries.
e The undertaking of researches and translations at hourly rates.
f A bulletin board for new books, and coming meetings and congresses.

Respectfully submitted

L. E. Howard,
Librarian

UNITED ENGINEERING SOCIETY

REPORT OF TREASURER

To the Board of Trustees
United Engineering Society

I beg to submit herewith report of the treasurer as of December 31, 1909.

From the balance sheet submitted herewith it appears that our physical property, over and above the value of the building and our equity in the land, consists of building equipment amounting to $16,767.72, and furniture and fixtures, $2,921.20.

During the current year there have been added to the real estate equipment account a toilet room on the twelfth floor, at an expense

of $530, and furniture and fixtures representing an expenditure of $682.86, including telephone booths, stereopticon, tables, chairs, etc.

It will be noted that the principal of the mortgage on the land, held by Andrew Carnegie, Esq., and amounting originally to $540,000, has been reduced by payments from the land and building funds of the Societies to $223,000, correspondingly reducing the burden on the Founder Societies for payment of interest.

The gross operating expenses for the year were $35,845.92 or, excluding expenditures for building equipment, $530, and furniture and fixtures to the amount of $682.86, a net cost of operating the building for the year 1908 of $32,163.57, slightly in excess of 1908.

In accordance with the resolution of the board an appropriation of $5000 was made out of the surplus remaining from the year 1908 and this amount ($5,037.50) was invested in Baltimore & Ohio bonds, as an addition to the contingency and renewal fund, as provided for in the Founders' Agreement, bringing the reserve fund up to $10,268.75. It is recommended that a similar appropriation be made out of the available balance from this year's operations, leaving a surplus to be carried forward of $3,905.95.

Attention is called to the fact that we had on January 24, 1910, unoccupied floor space in the building equivalent in rental value to only 4 per cent of the total space availab e for assessment and a part of this small remaining space is under option, so that the only space available and unengaged is the room and ante-room occupied by the Trustees as Board Room, and even that is occasionally called upon for board meetings under assessment for outside parties. One of the Founder Societies is prepared to release one or two rooms for applications from Associates, otherwise the building may be deemed fully occupied.

Your attention is directed to the small number of times the auditorium has been occupied during the year, thirty times as against twenty-seven times in 1908, and the relatively small demand for the two assembly rooms on the fifth floor, occupied fifty-six times in 1909 as against sixty-eight times in 1908. During the past year the facilities of the building were enjoyed by fifty-two societies, Founders and Associates, as against thirty-four in the year 1908. The limited use made of the auditorium and of the assembly rooms on the fifth floor, the income therefrom barely covering their quota of the fixed charges, continues to be a problem in the economical administration of the building.

The chief librarian reports a total attendance during the year of

8303 as against 7231 in 1908, the day attendance showing an increase of 750 and the evening attendance of 322.

The assessments paid for the year by the Founder Societies occupying one entire floor were $6000 each, representing a total expenditure by each, including interest on its full principal of mortgage on the land, of $13,000, reduced in each case to the extent the Society may have paid off part of its mortgage share. As the associate societies are assessed approximately $10,000 for equivalent facilities, it will be seen that the Founder Societies are still carrying more than their proportion of the carrying charges for equivalent office space occupancy in the building.

Respectfully submitted,

(*Signed*) J. W. Lieb, Jr.,

Treasurer

UNITED ENGINEERING SOCIETY

Balance Sheet, January 1, 1910

ASSETS

Real Estate, Land		$540,000.00	
Real Estate, Building		1,050,000.00	
Real Estate Equipment		16,767.72	
Furniture and Fixtures		2,921.20	
N. Y. City Bonds (cost) Reserve		5,231.25	
Balto. & Ohio Bonds (cost) Reserve		5,037.50	
Accounts Receivable		3,357.00	
Library United Engineering Society		29.05	
Library, adjustment account		30.56	
CASH			
Working Balance	$5,099.88		
For Reserve Fund	5,000.00		
Ways and Means Com.	1,165.08	11,264.96	
Petty Cash		500.00	
		$1,635,139.24	

LIABILITIES

Balance of Mortgage (Land) A.I.E.E.	$54,000.00	
Balance of Mortgage, (Land) A. S. M. E.	81,000.00	
Balance of Mortgage, (Land) A.I.M.E.	88,000.00	$223,000.00
A.I.E.E. Equity in Building		350,000.00
A.S.M.E. Equity in Building		350,000.00
A.I.M.E Equity in Building		350,000.00
A.I.E.E. Equity in Real Estate Equipment		3,346.61
A.S.M.E. Equity in Real Estate Equipment		3,346.62

A.I.M.E. Equity in Real Estate Equipment	3, 346.62
A.I.E.E. payments to date in liquidation of Mortgage on Land	126,000.00
A.S.M.E. payments to date in liquidation of Mortgage on Land	99,000.00
A.I.M.E. Payments to date in liquidation of Mortgage on Land	92,000.00
Depreciation and Reserve Fund	15,000.00
Ways and Means Committee, etc	1,165.08
Accounts Payable	1,150.00
Balance Cash, Accounts Received, Furniture, etc	17,784.31
	$1,635,139.24

Statement of Receipts and Disbursements Year Ending December 31, 1909

Cash

RECEIPTS

Balance on hand January 1, 1909	$6,510.63
Account Reduction of Mortgage on Land	64,000.00
Account Interest on Mortgage	10,740.00
Assessment of Founder Societies	18,000.03
Assessment of Associates, Offices, Meetings	27,363.29
Library Acount	5,109.27
Interest on Bonds	225.00
	$131,948.22

DISBURSEMENTS

Account Reduction of Mortgage on Land	$64,000.00
Account Interest on Mortgage	10,740.00
Operating Expense, Cash Expenditures	30,445.39
Real Estate Equipment	530.00
Furniture and Fixtures	682.86
Library Account	5,195.52
Bonds purchased (reserve)	5,037.50
Accounts Payable (from 1908)	1,399.37
A.I.M.E. return of rentals for office	660.00
Insurance	2,469.49
Library adjustment	688.21
Balance on hand, January 1, 1910	10,099.88
	$131,948.22

Operating Income and Expenses Year Ending December 31, 1909

INCOME

Assessment Founders	$18,000.03	
Less A.I.M.E. refund	660.00	$17,340.03
Assessment Associates		16,746.00
Assessment Miscellaneous (Offices and meetings)		5,991.50
Telephone returns		2,620.70
Miscellaneous charges to Societies		1,828.64
Interest		225.00
		$44,751.87

EXPENSES

Operating Expenses, gross	$32,163.57
Real Estate Equipment	530.00
Furniture & Fixtures	682.86
Reserve Fund	5,000.00
Insurance	2,469.49
Balance to surplus	3,905.95
	$44,751.87

NECROLOGY

CHARLES W. BATCHELOR

Charles W. Batchelor was born in London, England, December 21, 1845. Soon afterwards his parents moved to Manchester, where he received a liberal education and where he served his apprenticeship in several of the largest engineering works of that place. At the age of twenty-two years he came to this country to install some machinery for the Clark Thread Company of Newark, N. J., and almost from the first was for over twenty years intimately associated with the inventor Thomas A. Edison, assisting in the development of the electric pen, the telephone transmitter, the phonograph, the electric railroad and the Edison incandescent lamp and lighting system.

In 1881 he went to Europe to represent the Edison interests at the Paris electrical exhibition of that year, and remained in Paris, for three years where he was the first to introduce the system of electric lighting. He made the original installation at the Paris opera house, and started a number of isolated plants in other parts of Europe; at the same time establishing and managing a large factory at Ivry.

Returning to this country in 1884, he assumed the management of the Edison Machine Works, an organization which in course of time developed into the Edison General Electric Company, and the selection of the site of their large works at Schenectady was made by him. Later this company combined with the Thomson Houston Electric Company and became the General Electric Company.

Of late years he had practically retired from business and devoted much time to travel, though he retained the presidency of the Taylor & Co. iron foundry, a concern in which he had been interested since its establishment.

Mr. Batchelor was a member of the Natural History Museum of New York and the New York Botanical Garden. He was a member for a number of years of the American Geographical Society, the American Institute of Electrical Engineers and the American Electrochemical Society. He entered this Society in 1880.

He died at his residence in this city on January first, 1910.

PERCY A. SANGUINETTI

Percy A. Sanguinetti, one of the early members of the Society, was born in Kingston, Jamaica, B. W. I., June 17, 1844, and died at his home in Mt. Vernon, N. Y., on January 30, 1910.

At the age of 16, he entered service as an apprentice in the locomotive shops of his native town. A few years later he received an appointment to the British Navy Yards at Chatham, England, where he worked through the various departments. During this time he passed a successful examination as teacher of mechanical drawing in the evening mechanical schools at South Kensington, London. In 1867, he was appointed by the Admiralty Board to represent the town of Chatham at the Paris Exposition and to report upon its mechanical features.

His experience in the United States dates from the Centennial Exhibition at Philadelphia in 1876, where he served as assistant to the machinery bureau, designing the system of shafting and the cascade in the pump annex and assisting in the experiments with turbines. At the close of the Exhibition he entered the service of the Franklin Sugar Refinery in Philadelphia, where he remained twelve years, conducting during part of this time a course in mechanical engineering at Franklin Institute. In 1893 he acted as mechanical aid at the World's Columbian Exposition in Chicago and for the following three years occupied the chair of mechanical engineering at the Armour Institute of Technology. In 1895 he came to New York to engage in consulting practice and during the past two years has served in the appraisal bureau of the Public Service Commission.

In 1901, Mr. Sanguinetti secured the coöperation of a score of representative manufacturers of this country in the introduction of American machinery into Jamaica, especially in sugar plantation and power development. His latest work was the remodeling of a sugar refinery near New Orleans, which he completed just two months before his death.

APRIL 1910 VOL. 32 No. 4

THE JOURNAL
OF
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CONTENTS

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THE JOURNAL is published by The American Society of Mechanical Engineers twelve times a year, monthly except in July and August, semi-monthly in October and November.

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The professional papers contained in The Journal are published prior to the meetings at which they are to be presented, in order to afford members an opportunity to prepare any discussion which they may wish to present.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 APRIL 1910 NUMBER 4

THE St. Louis monthly meeting for April will be held Saturday evening, April 9, with the American Institute of Electrical Engineers and The Engineers' Club of St. Louis coöperating. The subject of the meeting will be The Electric Drive in the Machine Shop, with four papers constituting a comprehensive symposium, both from the standpoint of the mechanical equipment of machine tools and from the economic side of the saving to be effected by the installation of different types of motor drives.

Three of the papers are contributed by The American Society of Mechanical Engineers as follows: The Economy of the Electric Drive, by A. L. DeLeeuw, Mem. Am. Soc. M.E., published in The Journal for November 1909; The Economical Features of Electric Motor Applications by Charles Robbins, Assoc.Mem.A.I.E.E.; Mechanical Features of Electric Driving, by John Riddell, Mem. Am.Soc.M.E. The two latter papers are published in this number of The Journal.

The American Institute of Electrical Engineers have contributed a paper by Charles Fair upon Selection and Methods of Application of Motors and Controllers.

MEETING IN NEW YORK APRIL 12

The New York monthly meeting for April will be held in the Auditorium of the Engineering Societies' Building, Tuesday evening, April 12, the American Institute of Electrical Engineers coöperating. The subject of the The Electric Drive in the Machine Shop will be

discussed and the four papers previously mentioned in the announcement of the April St. Louis meeting will be presented by Messrs. A. L. De Leeuw, Charles Robbins, John Riddell and Charles Fair.

There will be discussions by Messrs. Gano Dunn, Vice-president of the Crocker-Wheeler Company, Ampere, N. J., Clarence L. Collens, President, Reliance Electric and Engineering Company, Cleveland, H. A. Hornor, Electrical Engineer, Philadelphia, Charles Day of Dodge & Day, Philadelphia, Henry Hess of Philadelphia, and other prominent engineers, among them an official representative of the Machine Tool Builders Association, who will report upon the efforts of this Association at standardization of electric motor equipment.

MEETING IN BOSTON APRIL 27

The Boston monthly meeting of the Society, April 27, in which the American Institute of Electrical Engineers, Boston Section, and the Boston Society of Civil Engineers, will coöperate, will be the occasion of the presentation of a paper by Prof. C. M. Allen of Worcester Polytechnic Institute, Mem. Am. Soc. M. E., on The Testing of Water Wheels after Installation, published in this issue of The Journal. The meeting will be held at 8 p.m., in the hall of the Edison Electric Illuminating Company, 39 Boylston St.

Preliminary announcement is also made of the May meeting, to be held May 18, in charge of the Boston Section of the American Institute of Electrical Engineers, with The American Society of Mechanical Engineers and the Boston Society of Civil Engineers coöperating. The gathering will be addressed by Lewis B. Stilwell, President Am. Inst. E. E., on the subject of conservation, especially of water power. The place of meeting will be announced by the American Institute of Electrical Engineers to members of the three societies.

MEETING IN NEW YORK, MARCH 8

The New York monthly meeting for March, in which the American Institute of Electrical Engineers coöperated, was the largest monthly meeting of the Society held during the present season. The paper by H. G. Stott, Mem. Am. Soc. M. E., and J. S. Piggott, on tests on the 15,000-kw. steam-engine-turbine unit located in the 59th Street Station of the Interborough Rapid Transit Company of New York, aroused an unusual amount of interest.

The attendance at the meeting was nearly 700, and the audience remained until a late hour to listen to the comments of those who wished to speak from the floor.

After the presentation by Mr. Stott of the main facts regarding these tests and the installation of the low-pressure turbine on which the tests were made, Mr. Piggott gave an interesting account of the methods used in conducting the tests, illustrating his talk with lantern slides.

A summary of the discussion at this meeting, together with an abstract of that given at the recent Boston meeting which Mr. Stott addressed on the subject of Low-pressure Turbines, and the topical discussion on Steam Turbines given at a meeting of the Society in St. Louis, will be published together in a subsequent issue.

MEETING IN BOSTON, MARCH 11

A meeting of the Society was held in Boston, March 11, in which the Boston Section of the American Institute of Electrical Engineers and the Boston Society of Civil Engineers coöperated. The total attendance was 125. Chairman Hollis said in his introductory remarks that in future all these gatherings would probably be meetings not of one society but of engineers in general, thus inaugurating a new movement among the engineering societies of the country of pretty thorough coöperation. As a result of the meeting on January 21, a committee of fifteen, representative of twelve of the thirty or forty different societies in the vicinity of Boston having especial scientific and engineering interests, had been formed and had under consideration the possibility of erecting an engineering building and perhaps organizing within it a club, as well as providing rooms for the various societies and a meeting place for such gatherings as this. This movement, he thought, would not only bring together the various societies but should serve to relate the engineering profession more closely to civic questions in Boston.

Magnus W. Alexander of Lynn, Mass., Mem.Am.Soc.M.E., presented his paper on The Training of Men, a Necessary Part of a Modern Factory System, published in The Journal for January. This was discussed by Henry E. Rhoades, Past Assistant Engineer, U. S. N., Retired, Prof. Chas. F. Park and R. H. Smith of the Massachusetts Institute of Technology, Prof. Ira N. Hollis, Mem. Am.Soc.M.E., Prof. Gardner C. Anthony, Mem.Am.Soc.M.E., Prof. Peter Schwamb, Mem.Am.Soc.M.E., Luther D. Burlingame, Mem.Am.Soc.M.E., G.

C. Ewing, Assoc.Am.Inst.E.E., S. Fred Smith, President, New England Section of the National Electric Light Association, H. S. Knowlton, Prof. Edward F. Miller, of the Massachusetts Institute of Technology, Mem.Am.Soc.M.E., and Dickerson G. Baker, Mem.Am.Soc.M.E.

MEETING IN ST. LOUIS, MARCH 12

A topical discussion was held by the Society and the Engineers' Club of St. Louis, in the club rooms on Olive Street, Saturday evening, March 12, on the subject of Flywheels. The discussion was opened by Dr. C. H. Benjamin, Mem.Am.Soc.M.E., Dean of the Schools of Engineering of Purdue University, who reviewed the recent developments in testing flywheels and gave an account of the latest apparatus and testing pit for this purpose at Purdue University. There were present at the meeting, Jesse M. Smith, Past-President, Col. E. D. Meier, Vice-President, and Calvin W. Rice, Secretary of this Society. An informal dinner was held before the meeting at the Mercantile Club, attended by the guests and local members.

As is customary when it is impossible for the president to be present, a vice-president is invited to preside, and Col. E. D. Meier, Vice-President, acted as chairman. The presiding officer and William H. Bryan, Chairman of the Local Committee, took the opportunity, previous to the topical discussion, to give the greetings of the Council to the members, and to explain the interest on the part of the Council in the meetings of the Society in the several cities and the appreciation by the Local Committee of their efforts. It was explained that all meetings of the Society are conducted under the same rules, wherever held, the national character of the Society being in this way developed; and at the same time the members are enabled to get the greatest benefits from their association.

Professor Benjamin, who has for many years conducted experiments on flywheels with a view to determining the bursting strength of different types of wheels, followed with his remarks on the subject, after which there was a general discussion.

SPRING MEETING ATLANTIC CITY

RAILROAD TRANSPORTATION NOTICE

Arrangements for hotel, transportation and Pullman car accommodations should be made personally.

Special concessions have been secured for members and guests attending the Spring Meeting in Atlantic City, May 31 to June 3, 1910.

The special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below. Read item *g*.

a Buy your ticket at full fare for the going journey, between May 27 and June 2 inclusive. At the same time request a certificate, *not a receipt*. This ticket and certificate should be secured at least half an hour before the departure of the train.

b Certificates are not kept at all stations. Ask your station agent whether he has certificates and through tickets. If not, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point, and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to S. Edgar Whitaker, office manager at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after June 3.

d An agent of the Trunk Line Association will validate certificates, June 1, 2, 3. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to June 8, on the same route over which the purchaser came, at three-fifths the rate.

g Members and guests from New York City should buy the regular round trip tickets at $5 and show the return portion to Mr. Whitaker or the validating agent.

The special rate is granted only for the following:

The Trunk Line Association:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

The Central Passenger Association:

The portion of Illinois south of a line from Chicago through Peoria to Keokuk and east of the Mississippi River, the States of Indiana, and Ohio, the portion of Pennsylvania and New York north and west of the Ohio River, Salamanca and Buffalo, and that portion of Michigan between Lakes Michigan and Huron.

The New England Passenger Association except via N. Y. O. & W. R. R. and Eastern Steamship Co. The Rutland R. R. participates in fares reading via its road:

Maine, New Hampshire, Vermont, Massachusetts, Rhode Island and Connecticut.

The Western Passenger Association offers revised one-way fares, on the basis of two cents per mile, to Chicago, Peoria and St. Louis; these three places are points in the Central Passenger Association, and from these points purchase round trip tickets, in the manner outlined in the preceding paragraphs:

North Dakota, South Dakota, Nebraska, Kansas, Colorado, east of a north and south line through Denver, Iowa, Minnesota, Wisconsin, Missouri, north of a line through Kansas, Jefferson City and St. Louis; Illinois, north of a line from Chicago through Peoria to Keokuk.

The Eastern Canadian Passenger Association:

Canadian territory east of and including Port Arthur; Sault Ste. Marie, Sarnia and Windsor, Ont.

MEETING IN GREAT BRITAIN

Invitations have been received through the Verein Deutscher Ingenieure, to the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, as well as the following industrial works:

INDUSTRIELLE WERKE

Accumulatorenfabrik A. G., Hagen i/Westif.
Düsseldorfer Eisenbahnbedarf vorm. Carl Weyer & Co., Düsseldorf Oberbilk-
Wegmann & Co., Kassel
Van der Zypen & Charlier, G.m.b.H., Cöln-Deutz
J. Goossens, Aachen (Eschweiler-Aue)
Eschweiler Bergwerksverein, Eschweiler
Gebr. Stumm, G.m.b.H., Neunkirchen (Bez. Trier)
Rheinisch-Nassauische Bergwerks- & Hütten Akt. Ges., Stolberg i/Rhld.
Ehrhardt & Sehmer, G.m.b.H., Schleifmühle, Post Saarbrücken
Dingler'sche Maschinenfabrik A. G., Zweibrücken (Pfalz)
Röchling'sche Eisen- und Stahlwerke, G.m.b.H., Völklingen (Saar)
Maschinen- und Armaturenfabrik vorm. Klein, Schanzlin & Becker, Frankenthal
Stahlwerke Mannheim, Rheinau bei Mannheim
Badische Aktiengesellschaft für Rheinschiffahrt und Seetransport, Mannheim
Heinrich Lanz, Lokomobilen und landwirtschaftliche Maschinen Mannheim (Firma bittet um namentliches Verzeichnis der Teilnehmer nebstAngabe der Firma und deren Branche. Ingenieure solcher Fabriken, die sich mit dem Bau von Lokomobilen und landwirtchaftlichen Maschinen beschäftigen, können an der Beschtigung nicht teilnehmen.)
Aktiengesellschaft vorm. Burgeff & Co., Schaumweinfabrik, Hochheim a/M.
L. Schuler, Werkzeugmaschinenfabrik, Göppingen
Aktiengesellschaft L. A. Riedinger, Maschinen-& Broncewarenfabrik, Augsburg
Maschinenfabrik Augsburg-Nürnberg, Aktiengesellschaft, Nürnberg
Siemens-Schuckertwerke, Nürnberg
Werkzeugmaschinenfabrik Union, vorm. Diehl, Chemnitz
J. E. Reinecker, Werkzeugmaschinenfabrik, Chemnitz-Gablenz
Dresdner Bohrmaschinenfabrik A. G. vorm. Bernhard Fischer & Winsch Dresden-A., Zwickauerstrasse 41-45
Aktiengesellschaft vorm. Gebr. Seck. Mühlenbauanstalt, Dresden-A.
Telefonfabrik Aktiengesellschaft vorm. J. Berliner, Hannover
Beuchelt & Co., Grünberg i/Schles.
A. Borsig, Tegel-Berlin
Siemens-Schuckertwerke, Berlin

Siemens & Halske, Aktiengesellschaft, Berlin
H. Aron, Elektrizitätszahlerfabrik, G.m.b.H., Berlin-Charlottenburg
Allgemeine Elektrizitätsgesellschaft, Berlin
Ludwig Loewe & Co., Aktiengesellschaft Berlin, NW Huttenstr. 17–20
R. Wolf, Lokomobilen, Magdeburg-Buckau
Deutsche Wagenbau-und Leihgesellschaft m.b.H., Danzig
Aktiengesellschaft Amme, Giesecke & Konegen, Mühlenbaustalt Braunschweig
Breslauer Aktiengesellschaft für Eisenbahn-Wagenbau, Breslau

WISSENSCHAFTLICHE INSTITUTE

Physikalisch-Tehnische Reichsanstalt, Charlottenburg-Berlin
Maschinenlaboratorium der Kgl. Techn. Hochschule, Berlin
Maschinentechnisches Laboratorium der Kgl. Techn. Hochschule Danzig
Maschinenlaboratorium der Kgl. Techn. Hochschule Aachen
Material prufüngsanstalt, Laboratorium und Institut der Grossherz. Techn. Hochschule Darmstadt
Materialprüfungsanstalt und Ingenierlaboratorium der Kgl. Techn. Hochschule Stuttgart
Grossherz. Badische chemisch-technische Prüfungs-und Versuchsanstalt, Lehr- und Versuchs-Gasanstalt und andere Institute an der Technischen Hochschule Karlsruhe
Königliches Material prüfungsamt Gross-Lichterfelde West. (Berlin)
Maschinenlaboratorium und elektrotechnisches Laboratorium der Kgl. Technischen Hochschule Breslau. (z. Zt. noch im Ausbau; da die Laboratorien im Juli ds. Jrs. vielleicht betriebsfähig sein können, empfiehlt sich eine vorherige Anfrage beim Rektorat der Hochschule.)

Translations of the invitations from the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, with the accompanying letter from the Verein deutscher Ingenieure, follow.

Members of the Society desiring to avail themselves of these courteous invitations will correspond with the Secretary that he may secure them letters of introduction.

VEREIN DEUTSCHER INGENIEURE
Berlin, N. W., Charlottenstrasse, 43

Col. E. D. Meier *Berlin, January 31, 1910*
Care of Heine Safety Boiler Co.,
11 Broadway, New York.

Referring to your correspondence with Dr. Ing. Diesel of Munich, concerning the inspection of German industrial works and scientific institutions by members of The American Society of Mechanical Engineers, we have the honor to send you a list of firms and institutions which have declared themselves ready to receive these gentlemen, and enclose also several letters.

As we are not informed of the traveling route of the members of your highly appreciated Society, and do not know their wishes or preferences, we consider

it advisable that The American Society of Mechanical Engineers should correspond directly with such works and institutions as they would like to visit and arrange the necessary details.

With highest regards,

Business office of the
VEREIN DEUTSCHER INGENIEURE
Linde

MASCHIHENBAULABORATORIUM DER KONIGL, TECHNISCHEN HOCHSCHULE ZU BERLIN

Vorsteher: Prof. E. Josse

Charlottenburg, November, 15, 1909.

Highly honored gentlemen:

I have been informed that on the occasion of your visit to England and the British Institution of Mechanical Engineers, you also think of visiting Germany in order to examine industrial works, scientific institutions, etc.

We would greet with pleasure your visit to the Königliche Technische Hochschule of Charlottenburg, and we would then gladly take the opportunity to show you the various laboratories of the Hochschule, especially the Mechanical Laboratory in all its details.

I beg you therefore to embody in your programme a visit to the above named institution and sign, with especial regards

Cordially,
JOSSE
Professor.

To The American Society of Mechanical Engineers.

MACHINENTECHNISCHES LABORATORIUM DER KGL. TECHNISCHE HOCHSCHULE
Prof. A. Wagener

Danzig-Langfuhr, November, 15, 1909.

To the American Society of Mechanical Engineers:

Having learned through the Verein deutscher Ingenieure that at the conclusion of the Annual Meeting of The American Society of Mechanical Engineers which is to take place about the end of July, 1910, in London, a visit to German industrial works and scientific institutions, is planned, I have the honor to extend a cordial invitation to The American Society of Mechanical Engineers to visit the mechanical laboratory of the Königliche Technische Hochschule at Danzig.

With highest regards

Respectfully
A. WAGENER.

SOCIETY AFFAIRS

MEETING OF THE COUNCIL

A regular meeting of the Council was held March 8, 1910, with E. D. Meier, Vice-President, presiding. There were present Charles Whiting Baker, George M. Bond, John R. Freeman, H. L. Gantt, James Hartness, Alex. C. Humphreys, F. R. Hutton, E. D. Meier, I. E. Moultrop, H. G. Reist, Jesse M. Smith, Arthur M. Waitt, and the Secretary. Regrets were received from President Westinghouse and W. F. M. Goss, Vice-President.

The minutes of the meeting of February 8 were read and approved, after being amended to read,

Voted: That the proposed amendments with the report of the Finance Committee, be referred to the Committee on Constitution and By-Laws jointly with the Finance Committee for report back to the Council.

The Secretary announced the deaths of A. M. Goodale and Percy A. Sanguinetti.

The resignations of A. E. Holcomb and H. J. Scales were accepted with regret. The membership of W. W. Bigelow was declared to have lapsed.

The Secretary read the invitation of President Aspinwall of the Institution of Mechanical Engineers, to members of the Council and Past Presidents of this Society, to a dinner on the evening of Monday, July 25. A total of 208 are planning to attend the Joint Meeting in England in July, of whom 135 have arranged to sail on the Celtic.

Letters were read from the Königliche Technische Hochschule of Berlin and of Danzig-Langfuhr, and the Verein deutscher Ingenieure extending invitations to the visiting members of the Society.

Voted: To appoint the following Committee on Land and Building Fund: H. F. Holloway, I. E. Moultrop, F. H. Stillman, Morris L. Cooke, George A. Orrok.

The Secretary stated that the memorial window to Sir Benjamin Baker, Hon. Mem.Am.Soc.M.E., deceased, was presented by the Earl of Cromer in behalf of the subscribers and accepted by the Dean of Westminster Abbey in December 1909, and that report including

a colored illustration of the window was in preparation by the Institution of Civil Engineers.

Resolutions were read from the members of the Society resident in St. Louis, together with the action of the Executive Committee, as follows:

Voted: That a committee to consist of Jesse M. Smith, Past-President, E. D. Meier, Vice-President, I. E. Moultrop, Manager, Willis E. Hall, Chairman of Meetings Committee, and Calvin W. Rice, Secretary, be appointed to represent the Council at the meeting of the Society in St. Louis, March 12.

The committee is instructed to present the greetings of the Council to the St. Louis members, to gather information regarding the meetings of the Society in St. Louis, and to report its findings to the Council together with its recommendations, as soon as practicable.

The communication of February 15 signed by Mr. Wm. H. Bryan, from the members of the Society in St. Louis, together with the communication of February 25 with respect to same from the Meetings Committee, was read; after consideration the Chairman of the Meetings Committee and the Secretary were requested to draft a letter for submission to the Executive Committee and the Council, which shall represent the views of the latter body.

The communication of February 23 from Mr. Wm. H. Bryan, representing the members of the Society in St. Louis, requesting permission to hold a meeting of the Society on March 12, referred by the Meetings Committee to the Executive Committee, was referred back to the Meetings Committee with power, with the suggestion that authorization be given for a topical discussion, namely, The Recent Developments in the Bursting of Flywheels and Pulleys, by Prof. C. H. Benjamin of Purdue University, the authorization to be given under By-Law 23.

Notice was given of the purpose of the Committee on Constitution and By-Laws to amend By-Laws 11, 16, 17, 18, as follows:

FEES AND DUES

B 16 The initiation fee and the annual dues for the first year shall be due and payable on the first day of the month following the date of the election of a Member, Associate or Junior. The annual dues for each ensuing year shall be due and payable in advance on the corresponding day in each year thereafter. Upon the payment of the initiation fee and the annual dues for the first year, the person elected shall be entitled to the rights and privileges of membership in the grade to which he was elected. The date of payment of a member's annual dues may be changed to the first day of any other month, and a *pro-rata* adjustment of the dues made, by application to the secretary.

B 17 A Member, Associate or Junior in arrears for dues for one year, on the first day of October previous to the Annual Meeting, shall not be entitled to

vote, or to receive the Transactions or the publications issued by the Society thereafter until such dues have been paid. Should the arrears for dues or otherwise be for more than two years, the name of such person shall be presented to the Council for such action as it deems advisable under C 24. Should the right to vote, or to receive the publications of the Society be questioned, the books of the Society shall be conclusive evidence.

B 18 The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

ELECTION OF MEMBERS

B 11 Each person elected to membership, except an Honorary Member must subscribe to the Constitution, By-Laws, and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society. Resignations from membership shall be presented to the Council for action.

It was also voted to approve the adoption of B 22 and B 24, as published in another part of this issue of The Journal.

Voted: To amend Rule 24 to read as follows:

R 24 Engineers and others not members of the American Society, but desiring to participate in the meeting of the Section, may enroll themselves as affiliates as heretofore provided with the approval of the Executive of the section. Such affiliates shall have the privilege of presenting papers and taking part in the discussions. They shall pay $5 per annum which shall be due and payable in advance, on October 1 of each year of their enrollment, and shall thereby be entitled to receive the regular issues of The Journal for a period covered by their dues.

Voted: To amend Rule 29 to read:

R 29 The American Society of Mechanical Engineers will furnish monthly issues of The Journal to all members of affiliated organizations who are not members of The American Society of Mechanical Engineers upon the payment by each of two dollars per year, such payment being due January 1 of each year. The American Society of Mechanical Engineers will furnish gratis to each affiliated body, extra copies of advance papers for use at its meetings, the number furnished to be agreed upon at the discretion of the Secretary.

Voted: To approve the report of the Committee on International Standard for Pipe Threads.

Voted: To refer the communication of John Riddell, suggesting the preservation of models of epoch-making inventions that would be of interest in showing the development of engineering, to a committee of five, consisting of Charles Wallace Hunt, Col. E. A. Stevens, John Riddell, F. R. Hutton, Edward Van Winkle, to report what action if any, should in their judgment be taken. The letter is appended to this report.

Voted: To approve the recommendation of the Executive Committee, with regard to Col. R. S. Crompton's communication respecting the desirability of an international standard for machine screws; that the Secretary be directed to reply that inasmuch as the Society is to meet in England this summer the Institution would possibly be pleased to invite the Society to a conference on this subject; if so, our Society would be pleased to respond by appointing a committee to confer with a committee of the Institution. It was also advised that a suggestion be included regarding the coöperation of the Automobile Engineers of America.

A communication was read from the Verein Deutscher Ingenieure, inviting the coöperation of the Society in securing biographies and reminiscences of our engineers, and it was voted to refer this to the Committee on Society History.

A letter was read from L. B. Stillwell, President of the American Institute of Electrical Engineers, expressing cordial coöperation in the meeting of the Society on April 12.

The meeting adjourned to April 12.

Amendments to B 22 and B 24 Approved by Council

The Council voted to approve under the provision of C 59 the adoption of B 22 and B 24. The Amendment is to take effect immediately, as provided.

FINANCE COMMITTEE

B 22 The Finance Committee shall consist of five Members or Associates. The term of office of one Member of the Committee shall expire at the end of each Annual Meeting. This Committee shall under the direction of the Council, have a supervision of the financial affairs of the society, including the books of account. The Committee may cause the accounts of the Society to be audited and approved annually by a chartered or other competent accountant. The Committee shall hold monthly meetings for the audit of bills and such other business as shall come before it and shall deliver to the Secretary for representation to the Council at the end of each fiscal year, a report of the financial condition of the Society for the past year, and also shall present therewith a detailed estimate of the probable income and expenditure of the Society for the following twelve months. It shall make recommendations to the Council as to investments, and when called upon by the Council, advise upon financial questions. It shall have charge of the making of all contracts and other obligations to pay money in the Society's work and the ordering of all expenditures thereunder.

PUBLICATION COMMITTEE

B 24 The Publication Committee shall consist of five Members or Associates. The term of office of one Member shall expire at the end of each Annual Meeting. The Committee shall review all papers and discussions which have been presented at the meetings, and shall decide what papers and discussions, or parts of the same, shall be printed in the Transactions of the Society. The Committes shall have the supervision of the monthly publication of the Society known ae "The Journal." The Committee will be expected to publish all such data as will be of assistance to engineers or investigators in their work. At the end of each fiscal year the Committee shall deliver to the Secretary for presentation to the Council a detailed report of its work.

Proposed Museum of Mechanic Arts

The communication previously referred to, by John Riddell, proposing a museum of Mechanic Arts, is as follows:

February 26, 1910

To the President and Council
The American Society of Mechanical Engineers,
29 West 39th Street, New York City.

Gentlemen:

The rapid advancement in Mechanic Arts in this country causes machinery and mechanical devices to be quickly superseded by newer developments. Such superseded devices are often most interesting, in studying the development of engineering, but unfortunately such devices disappear, leaving no record of what has been done. Think for a moment how interesting it would be to have the original and a few intervening forms of such American devices as the cotton gin, and a few of the numerous machines which we have made for harvesting cotton, the sewing machine, telephone, phonograph, linotype, telegraph, aeroplane, and other machines, while not of American origin, but to which we have greatly contributed in their development:—as the steam engine, plow, machine tools, general automatic machinery, bicycle, automobile, steam turbine, electric dynamo, and steel mill machinery.

In the South Kensington Museum the epoch-making inventions of Watt and Stephenson are carefully preserved, and visited by thousands of people from all civilized countries.

American Engineers have contributed so much to this work that it is appropriate that a record of the most important developments should be kept. I have thought for some time that a movement should be put on foot to establish a museum of mechanic arts, and that it is most appropriate that this should be done by The American Society of Mechanical Engineers.

It is important if anything is to be done in this line that action should not be delayed, since it will be readily understood that much material is available for such a museum now that could not be had ten years hence.

I am prompted to make this suggestion in view of the forthcoming trip abroad by many of your members and I might also suggest that some investigations ad report be made on the feasibility of such a scheme.

Yours truly

(*Signed*) JOHN RIDDELL.

Members who think a museum of such a character would be desirable are urged to make suggestions in regard to its scope as well as of articles that should be so preserved, or to manifest their interest by assisting to endow the museum or in any other manner.

STUDENT BRANCHES

BROOKLYN POLYTECHNIC INSTITUTE

William Kent, editor of Industrial Engineering and member of the Society, will give an address on Engineering and Common Sense, at the regular meeting of the Brooklyn Polytechnic Student Section, on Saturday evening, April 9, 1910, in the institute chapel, 85 Livingston St., Brooklyn.

PURDUE UNIVERSITY

On March 3, 1910, the Purdue Student Section was addressed by G. A. Weschler (1910), on The Manufacture of Cartridge Cases for the United States Navy. Mr. Weschler's eight years' experience in the Washington navy yard, two years of which were spent in the cartridge case shop, has made him familiar with a manufacturing process not generally known. The various steps in the manufacture were taken up in detail, the talk being illustrated by lantern slides.

STEVENS INSTITUTE OF TECHNOLOGY

The following lectures, of the schedule for 1909-1910, will be delivered before the Stevens Engineering Society: April 5, The Theory of Gyrostatic Motion, by Lewis A. Martin; April 12, Methods and System in Relation to Handling Concrete Work, by Frank B. Gilbreth, Mem.Am.Soc.M.E.; April 19, Notable Examples in Modern Construction, by John C. Ostrup; April 26, Development of the NewNavy, by David Watson Taylor; May 10, The Contribution of Photography to our Knowledge of the Stellar Universe, John A. Brashear, Hon. Mem. Am. Soc.M. E.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

The speakers at the annual meeting and banquet of the Mechanical Engineering Society of the Massachusetts Institute of Technology, were Prof. Gaetano Lanza, Mem.Am.Soc.M.E., Honorary Chairman of the organization, Colonel Locke, and C. C. Pierce. The following officers were elected for the ensuing year: Morril MacKenzie, Chairman; H. C. Brown, Vice-Chairman; Foster Russell, Secretary; H. S. Lord, Treasurer; D. P. Allen, H. S. Smith and A. F. Kenrick, Governing Committee.

NECROLOGY

JAMES HENRY BLESSING

James Henry Blessing was born in the village of French's Mills in Albany County, New York, September 14, 1837. At his father's death in 1849 he left school and in 1853 was apprenticed to the machinist trade for four years, with the firm of F. & T. Townsend. He remained with this firm until 1861, when he entered the U. S. Navy as acting assistant engineer. After the war he became engineer in charge of steam machinery of the Brooklyn Horse R. R. Co., returning to Albany in 1868 to act as superintendent of the foundry and machine works of Townsend & Jackson, successors to F. & T. Townsend.

In 1870 Mr. Blessing invented the return steam trap, the best known of his one hundred and twenty inventions. In July 1872 he left the employ of Townsend & Jackson to engage with Genl. Frederick Townsend in the manufacture and sale of these and other steam specialties, under the firm name of Townsend & Blessing. In June 1873 this firm sold their interest to the Albany Steam Trap Company and Mr. Blessing became secretary and treasurer and general superintendent of the company, and afterwards president.

Mr. Blessing was elected mayor of Albany in 1899. He was a member of the Society of Engineers of Eastern New York and of the Albany Historical and Art Society, and entered this Society in 1891. He died in Albany, February 21, 1910.

ALFRED MONTGOMERY GOODALE

Alfred Montgomery Goodale, Manager of the Society, 1898–1901, and a life member, died in Waltham, Mass., December 17, 1909.

He was born in Saco, Me., December 20, 1855, and educated in the public schools of that State, receiving from the Maine State College the degree of B.S. in 1875. He served for five years in the works of the Saco Water Power Machine Shop, Biddeford, Me., and the Bates Mills, Lewiston, Me., building, setting up and running cotton machinery; and in 1880 became superintendent of the Newton Mills, at New-

ton Upper Falls, Mass., afterwards acting as agent, from 1881 to 1883, and from 1884 to 1894, for the Hamilton Woolen Company of Amesbury, Mass., and the Boston Mfg. Co., of Waltham, Mass. Since 1894 he had been treasurer and a director of the Boston Mfg. Co. With each of these companies, Mr. Goodale had charge of the erection of new machinery, engines and boilers, and of the reorganization and improvements of the existing plants. In 1901 he started the firm of A. M. Goodale & Co., of Boston, brokers in cloths and yarns.

Mr. Goodale served on various city commissions and was a trustee of the Waltham hospital. He was a director of the New England and Northwestern Investment Company and of the Westfield Creel Company. Besides his club and Masonic connections he was president of the New England Cotton Manufacturers Association, and a member of this Society since 1886.

J. HENRY SIRICH, JR.

J. Henry Sirich, Jr., associate member of the Society, died January 22, 1910, at Bethlehem, Pa. He was born at Baltimore, Md., April 9, 1881, and received his technical education at the Baltimore Polytechnic Institute, from which he was graduated in the class of 1898. He served his apprenticeship at and afterwards entered the drafting room of the engineering works of Robert Poole & Son Company, now the Poole Engineering & Machine Company of Baltimore.

In April 1903 Mr. Sirich became an assistant engineer on the steamships of the Atlantic Transport Company, remaining with them a year, during which he secured a United States license as second assistant engineer of ocean condensing steamers of 10,000 gross tons. From 1904 to 1908 he was connected with the American Bridge Company at Ambridge, Pa., and the Westinghouse Machine Company, East Pittsburg, Pa. In the latter company he entered first the turbine-testing department, of which he became foreman in September 1905, and later was transferred to the turbine-erecting department of the New York district, as trouble foreman.

In July 1908 he became connected with the power department of the Bethlehem Steel Company and held the position of chief draftsman at the time of his death. Mr. Sirich was accidentally drowned in the Lehigh River at Bethlehem, Pa.

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OF

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CONTENTS—Continued

The professional papers contained in The Journal are published prior to the meetings at which they are to be presented, in order to afford members anopportunity to prepare any discussion which they may wish to present.

THE JOURNAL

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 MAY 1910 NUMBER 5

SPRING MEETING, ATLANTIC CITY, MAY 31-JUNE 3

PROGRAM

Tuesday afternoon and evening, May 31

Informal reunion of members in the parlors of the Marlborough-Blenheim.

Wednesday, June 1, 10 a.m.

PROFESSIONAL SESSION

Business meeting. Reports of Committees, Tellers of Election, New Business.

PAPERS ON MACHINE CONSTRUCTION AND OPERATION

The Shockless Jarring Machine, Wilfred Lewis.

A Comparison of Lathe Headstock Characteristics, Prof. Walter Rautenstrauch.

The Strength of Punch and Riveter Frames made of Cast Iron, Prof. A. L. Jenkins.

Wednesday afternoon and evening

The afternoon is left unassigned to give opportunity for sight-seeing. Roller chairs for the board walk will be available for the visiting members and guests through the courtesy of the Local Committee.

In the evening, entertainment on the steel pier has been provided by the committee.

Thursday, June 2, 10 a. m.

PROFESSIONAL SESSION

MISCELLANEOUS PAPERS

The Mechanical Engineer and the Textile Industry, H. L. Gantt.

The Elastic Limit of Manganese and other Bronzes, J. A. Capp.

The Hydrostatic Chord, R. D. Johnson.

The Resistance of Freight Trains, Prof. Edw. C. Schmidt.

Thursday, 2 p. m.

GAS POWER SECTION

BUSINESS MEETING AND REPORTS OF COMMITTEES.

PAPERS

A Regenerator Cycle for Gas Engines Using Sub-Adiabatic Expansion, Prof. A. J. Frith.

Gas Engines for Driving Alternating Current Generators, H. G. Reist.

Two Proposed Units of Power, Prof. Wm. T. Magruder.

Some Operating Experiences with a Blast Furnace Gas Power Plant, H. J. Freyn.

Thursday, 9 p. m.

Reception, followed by conferring of Honorary Membership on Rear-Admiral George W. Melville, U. S. N., Ret. A brief address will be made by Admiral Melville, and the evening will conclude with dancing and refreshments.

Friday, June 3, 10 a. m.

PROFESSIONAL SESSION

PAPERS ON POWER TRANSMISSION

Improvements in Lineshaft Hangers and Bearings, Henry Hess.

Experimental Analysis of a Friction Clutch-Coupling. Prof. Wm. T. Magruder.

An Improved Absorption Dynamometer, Prof. C. M. Garland.

Critical Speed Calculation, S. H. Weaver.

REPRESENTATIVE OF THE PROFESSION IN PENNSYLVANIA AND NEW JERSEY

For several years the Spring Meeting of the Society has been held in cities where there has been an opportunity of visiting places of interest and inspecting engineering enterprises, and the time of the members has been very fully occupied in taking advantage of such excursions as the generosity and coöperation of the local members have made possible. While these meetings have all been thoroughly enjoyed, it was thought that it would be a welcome change to hold a meeting at a resort where those attending would have more time for renewal of acquaintance and for personal intercourse, instead of devoting so much attention to matters outside of the interests directly related to the Society and its membership. The last meeting of this sort was the Spring Meeting in 1903 held at Saratoga. The present meeting at Atlantic City should be an equally pleasant occasion, since there is no place in the country better adapted for holding a convention and the meeting is held at a time which is one of the most delightful in which to spend a few days on the New Jersey shore.

The Marlborough-Blenheim Hotel, the Convention headquarters, is situated at the central point of Atlantic City's famous seven-mile board walk, and occupies a block and a half on the ocean front looking southward and 200 yards on the City Park looking eastward. It has a capacity of 1100 and makes many provisions for the comfort of its guests, including the open-air plaza and the solariums overlooking the ocean.

LOCAL COMMITTEE

James M. Dodge, *Chairman*

J. Sellers Bancroft	Edward I. H. Howell	T. F. Salter
J. C. Brooks	Arthur C. Jackson	Coleman Sellers, Jr.
James Christie	William C. Kerr	Oberlin Smith
Morris L. Cooke	Wilfred Lewis	H. W. Spangler
Charles Day	E. P. Linch	A. A. Stevenson
Kern Dodge	Thomas C. McBride	Fred. W. Taylor
Francis H. Easby	D. T. MacLeod	J. A. C. L. de Trampe
Thomas M. Eynon	Edgar Marburg	Wm. S. Twining
John Fritz	Geo. W. Melville	Mr. Van Gilder
Harris R. Greene	Edwin A. Moore	S. M. Vauclain
G. T. Gwilliam	Henry G. Morris	William R. Webster
E. P. Haines	John S. Muckle	Tilden White
Robert E. Hall	John C. Parker	Walter Wood
Henry Hess	F. R. Pleasonton	

MEETING IN ENGLAND

A program of the joint meeting of The American Society of Mechanical Engineers and the Institution of Mechanical Engineers has been issued by the Institution. As already announced this meeting will be held in Birmingham and London and will begin on Monday, July 25. A local committee consisting of the Right Hon. the LORD MAYOR OF BIRMINGHAM, Alderman W. H. Bowater, together with members of the Institution and other gentlemen resident in the neighborhood, has been formed to make the necessary arrangements. A ladies' committee will be formed in Birmingham to make arrangements for the entertainment of ladies accompanying the members of both societies.

PROVISIONAL BIRMINGHAM PROGRAM

Monday, 25th July. Arrival in Birmingham

Tuesday, 26th July

Morning.—The Right Hon. the Lord Mayor of Birmingham and the Members of the Local Committee will receive and welcome the President, GEORGE WESTINGHOUSE, Esq., and the Officers and Members of the American Society of Mechanical Engineers, and the President, JOHN A. F. ASPINWALL, Esq., and the Council and Members of the Institution of Mechanical Engineers.
READING AND DISCUSSION OF PAPERS.
LUNCHEON in the Town Hall.

Afternoon.—Visits to Stratford-on-Avon, Worcester, Gloucester, or Bournville; and local Works.

Evening.—Garden Fête.

Wednesday, 27th July

Morning.—READING AND DISCUSSION OF PAPERS.
LUNCHEON in the Town Hall.

Afternoon.—Visits to the University and local Works.

Evening.—RECEPTION in the Council House, by invitation of the Right Hon. the Lord Mayor of Birmingham.

Thursday, 28th July

Visits to Works in Coventry and Rugby; also to Warwick, Leamington, Kenilworth, or Lichfield.

PROVISIONAL LONDON PROGRAM

Thursday, 28th July

Evening.—Conversazione at the Institution.

Friday, 29th July

Morning.—READING AND DISCUSSION OF PAPERS.
Afternoon.—Garden Parties at Private Houses.
Evening.—INSTITUTION DINNER in the Connaught Rooms, Freemason's Hall, Great Queen Street, W. C. (Including Ladies.)

Saturday, 30th July

Morning and Afternoon.—Excursion by Rail and River to WINDSOR and HENLEY.
Evening.—Reception at the Garden Club in the Japan-British Exhibition at the White City.

It is intended that Invitation Cards be handed to the American visitors on their arrival in Birmingham.

The privileges and invitations in connection with the Meeting are *personal* and are *not transferable.*

PRINCIPAL HOTELS CENTRALLY SITUATED IN BIRMINGHAM AND NEIGHBORHOOD

Queen's
Grand
Imperial
Midland
Colonnade
Swan
Plough and Harrow (Hagley Road $1\frac{1}{2}$ miles)
Cobden (Temperance)
Hen and Chickens (Temperance)
Kenilworth (27 miles): King's Head; Manor House
Knowle (10 miles): The Forest
Leamington ($23\frac{1}{2}$ miles): Regent; Clarendon: Manor House
Lichfield (18 miles): George; Swan
Stratford ($26\frac{1}{2}$ miles): Shakespeare; Red Horse; Red Lion
Warwick (22 miles): Warwick Arms
Wolverhampton ($12\frac{1}{2}$ miles): Victoria Star and Garter.

For the convenience of members of The American Society of Mechanical Engineers, who are expected to arrive in Liverpool on Sunday, 24th July, and to proceed by special train to Birmingham on the 25th, a list of the principal hotels in Liverpool and the neighborhood, and in Southport, follows.

PRINCIPAL HOTELS IN LIVERPOOL AND NEIGHBORHOOD

Adelphi, Ranelagh Place
Exchange Station, Tithebarn Street
North Western, Lime Street
Hotel St. George, Lime Street
Angel, Dale Street
Compton, Church Street
Feathers, Clayton Square
Imperial, Lime Street
Stork, Queen Square
Union, Parker Street
Washington, Lime Street
Waterloo, Clayton Square
Laurence's Temperance, Clayton Square
Shaftsbury (Temperance), Mount Pleasant.
Hotel Victoria, New Brighton ($2\frac{1}{2}$ miles by ferry boat)
Leasowe Castle Hydro., Wallasey ($3\frac{3}{4}$ miles)
Royal, Waterloo (5 miles)
Blundell Sands, Blundellsands (6 miles)
Royal, Hoylake ($7\frac{1}{4}$ miles)
New Hydro., West Kirby ($8\frac{1}{2}$ miles)
Chester (15 miles from Birkenhead: Queen Hotel (opposite Railway Station); Grosvenor Hotel (center of City); Blossoms Hotel
Southport ($18\frac{1}{2}$ miles from Liverpool): Prince of Wales. Lord Street; Palace; Royal; Victoria; Waverley; Queen's, all on Promenade.

MEETING IN ST. LOUIS MAY 14

A meeting of the Society will be held in St. Louis, May 14, in which the Engineers Club of St. Louis are to coöperate. The paper will be Freight Train Resistance by Prof. Edward C. Schmidt, which is published in this number of the Journal.

REPORTS

MEETING IN ST. LOUIS APRIL 9

The meeting of the engineers of St. Louis, April 9, was conducted by The American Society of Mechanical Engineers with the coöperation of the St. Louis Section of the American Institute of Electrical Engineers as well as of the Engineers Club of St. Louis A symposium on Electric Drive in the Machine Shop was presented, to which three papers were contributed by the Society: The Economy of the Electric Drive, by A. L. DeLeeuw, Mem. Am. Soc. M. E.; Economical Features of Electric Motor Applications by Charles Robbins, of the Westinghouse Electric and Manufacturing Company, and associate member of the American Institute of Electrical Engineers; Mechanical Features of Electric Driving, by John Riddell, Mem. Am. Soc. M. E. A paper, Selection and Methods of Application of Motors and Controllers, by Charles Fair, of the General Electric Company, a member of the Institute was contributed by the American Institute of Electrical Engineers. The attendance was nearly 100.

MEETING IN NEW YORK APRIL 12

The New York monthly meeting was held Tuesday evening, April 12, in the Auditorium of the Engineering Societies Building, with the American Institute of Electrical Engineers coöperating. The subject was Electric Drive in the Machine Shop with the four papers listed above under the meeting at St. Louis.

This subject of electric driving has long been in preparation with a view to presenting in the papers and discussions the recent developments in electric motor applications to machine tools and the economic features of such applications where motors are installed either for direct driving or in connection with lineshafts for group driving. The economic side was very fully discussed by representatives of the machine tool industry ando ther users of motors as well as by the motor manufacturers. Mr. Fred. L. Eberhardt, Vice-President of the National Machine Tool Builders Association, spoke officially for his

organization of their efforts, with the American Association of Electric Motor Manufacturers, for the securing of standards for motor equipment. Mr. A. L. DeLeeuw, a member of the committee of the National Machine Tool Builders Association to consider this subject, followed with a detailed account of the efforts at standardization to date, in which he said fifteen points had been raised for discussion and that seven of them had been considered by the joint committee thus far, among them being the subjects of horsepower, voltages, speeds and ratings.

The papers were also discussed by Henry Hess, of the Hess-Bright Manufacturing Company, Philadelphia, Pa.; L. R. Pomeroy, of the Safety Car Heating & Lighting Company, New York; Gano Dunn, Vice-President of the Crocker-Wheeler Company, Ampere, N. J.; Charles Day, of Dodge & Day, Philadelphia; W. S. Rogers of the Bantam Anti-Friction Company, Bantam, Conn.; Carl G. Barth, Philadelphia; and H. A. Horner, Philadelphia.

MEETING OF THE COUNCIL

A meeting of the Council was called to order in the rooms of the Society, April 12, 1910. Present, Messrs. George M. Bond, Chas, Whiting Baker, J. Sellers Bancroft, H. L. Gantt, James Hartness, Charles Wallace Hunt, F. R. Hutton, I. E. Moultrop, E. D. Meier, H. G. Reist, Frederick W. Taylor, Jesse M. Smith and the Secretary. In the absence of the President, Col. E. D. Meier took the chair.

The Secretary announced the deaths of James Blessing and Gardiner C. Sims.

The resignations of A. E. Coleman, Jr., Zareh H. Kevorkian, George E. Kirk, J. E. Tatnall and Ephraim Smith were read and accepted, and the membership of Thomas M. Keith, George L. Holmes, Barton H. Cameron, Rafael de la Mora, W. Allen Pendry, Edward S. Seaver and Charles L. Weil was declared to have lapsed.

Voted: To accept the report of the special committee appointed by the Council to go to St. Louis, and to express the very hearty appreciation of the Council; and to refer the report to the Executive Committee.

The Executive Committee reported as the total booking on the Celtic, to date, of those who will attend the Joint Meeting in England: 144 members and ladies; going by other routes or already in Europe 81 making a grand total of 225.

Voted: To appoint as a Committee on Arrangements, in con-

nection with the joint meeting in England, Ambrose Swasey, *Chairman*, Charles Whiting Baker, *Vice-Chairman*, Dr. W. F. M. Goss, George M. Brill, John R. Freeman, and, ex-officio George Westinghouse, President, William H. Wiley, Treasurer, F. R. Hutton, Honorary Secretary, Willis E. Hall, Chairman Meetings Committee, and Calvin W. Rice, Secretary.

The Secretary reported that fifteen members of the Council and Past-Presidents expected to attend the dinner to by given by President Aspinwall, on the evening of Monday, July, 25.

Voted: That Charles Whiting Baker be appointed Honorary Vice-President, to represent the Society at the International Congress of Mining Metallurgy, Applied Mechanics and Practical Geology. The Secretary also presented to Mr. Baker the appointment from the State Department as delegate from the United States.

Voted: To appoint Worcester R. Warner Chairman of the Committee on Land Fund.

Voted: To refer to the Executive Committee, with power, in the matter of coöperation with the Verein Deutscher Ingenieure in the preparation of biographies of eminent engineers.

Voted: To approve the applications for a Student Branch at the University of Arkansas, Fayetteville, Ark.

Voted: To adopt the following amendments:

ELECTION OF MEMBERS

B 11 Each person elected to membership, except an Honorary Member must subscribe to the Constitution, By-Laws, and Rules of the Society, and pay the initiation fee before he can receive a certificate of membership in the Society. Resignations from membership shall be presented to the Council for action.

FEES AND DUES

B 16 The initiation fee and the annual dues for the first year shall be due and payble on the first day of the month following the date of the election of a Member, Associate, or Junior. The annual dues for each ensuing year shall be due and payable in advance on the corresponding day in each year thereafter.

Upon the payment of the initiation fee and the annual dues for the first year the person elected shall be entitled to the rights and privileges of membership in the grade to which he was elected. The date of payment of a member's annual dues may be changed to the first day of any other month, and a *pro-rata* adjustment of the dues made, by application to the Secretary.

B17 A Member, Associate or Junior in arrears for dues for one year, on the first day of October previous to the annual Meeting, shall not be entitled to vote, or to receive the transactions or the publications issued by the Society

thereafter until such dues have been paid. Should the arrears for dues or otherwise be for more than two years, the name of such person shall be presented to the Council for such action as it deems advisable under C 24. Should the right to vote, or to receive the publications of the Society be questioned, the books of the Society shall be conclusive evidence.

B 18 The council may, in its discretion, restore to membership any person dropped from the roll for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

Voted: That the Council accept the invitation of the National Steam and Hot Water Fitters Association to a conference leading to the adoption of uniform standards for flanged and screwed cast-iron fittings, and that the Secretary communicate with the various members of the Council and specialists in power-house practice and ask their suggestions for names for such a committee.

Voted: That a message of congratulation and greeting be sent to the Aero Club of American on the opening of the club rooms on the evening of Wednesday, April 13, in the Engineering Societies Building.

Voted: That Charles Whiting Baker be appointed Chairman of a Committee, with power to increase the number to investigate the matter of a proposed bill now before the legislature to license engineers, and to report to the Council at its next meeting.

REQUEST FOR 1903 YEAR BOOK

A copy of the Year Book for 1903 is needed to complete the files of the Society. Any member willing to furnish a copy will please communicate with Calvin W. Rice, Secretary, at the rooms of the Society.

STUDENT BRANCHES

PENNSYLVANIA STATE COLLEGE

At the March meeting of the section held March 16, the topic for the evening was Methods of Coal Mining, which was ably handled by George B. Wharen, A. F. Goyne and Roy B. Fehr (1910). The papers were supplemented by views of mines and mine apparatus thrown on the screen. At the April meeting, Refrigeration and Cold Storage will be discussed.

PURDUE UNIVERSITY

On March 24, F. H. Clark, Genl. Supt. M.P. of the C. B. & Q. R. R. addressed the student section of Purdue University on The Functions and Work of the Motive Power Department, followed by a general discussion in which many points of interest were enlarged upon. Professor Ensley, of the university, addressed the meeting on April 6, on Recent Developments in Brake Shoe Tests, on whichhe is an authority. His address was supplemented by lantern slides.

UNIVERSITY OF CINCINNATI

The student branch of the University of Cincinnati had as the speaker at its meeting on March 25, James B. Stanwood, Mem.Am. Soc.M.E., who presented a very interesting paper on The Development of Non-Condensing Engines. Harry M. Lane, Mem.Am.Soc. M.E., gave a discussion of the paper. At the meeting on April 15, William Goodman, manager of Laidlaw-Dunn-Gordon Company, presented a paper on Air Compressors and their Manufacture.

WISCONSIN UNIVERSITY

At the April meeting of the section, G. A. Glick (1910) presented a paper on A 15,000-kw. Steam Engine Turbine Plant, based on Mr. Stott's paper published in The Journal, March issue. The paper was followed by a discussion in which Assistant Professor A. G. Christie told of some of the difficulties encountered in testing the engines referred to in the paper. The following officers were elected; President, John S. Langwill; Vice-President, Henry A. Christie; Corresponding Secretary, Karl L. Kraatz; Assistant Secretary, Guy H. Suhs; Treasurer, Angus MacArthur.

OTHER SOCIETIES

INTERNATIONAL CONGRESS OF MINING, METALLURGY, APPLIED MECHANICS AND PRACTICAL GEOLOGY

Charles Whiting Baker, Vice-President of the Society, has been appointed Honorary Vice-President to represent the Society at the International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, to be held at Düsseldorf, Germany June 20–23, 1910. Mr. Baker has also received appointment from the State Department as delegate from the United States.

BOSTON SOCIETY OF ARCHITECTS

At the dinner of the Boston Society of Architects, held at the Parker House, April 1, 1910, Calvin W. Rice, Secretary Am. Soc.M.E., was the guest of honor. The topic for consideration was Office Organization. Mr. Shreave of Carrere & Hastings was also a guest and spoke from knowledge not only of the office with which he is connected, but also of that of McKim, Mead & White.

Mr. Rice took occasion to explain the necessity of organization in the office of the society, as in a business, for the reason that members of the society who are business men expect efficient service whenever they communicate with the society on any matter. The Society is essentially an organization of trained men. The variety of the inquiries, several hundred a day, also requires a complete staff, and it must be organized if useful and practical attention is to be given these inquiries. The most important idea in connection with an engineering society is that, in a larger sense than the individual, it must serve the profession; rather than that it is simply an aggregation of persons for selfish interests. In other words, the association must be organized for progressive and helpful work. Such is the organization of The American Society of Mechanical Engineers.

AMERICAN ELECTROCHEMICAL SOCIETY

In response to a request, the Secretary has sent to Dr. Jos. W. Richards, South Bethlehem, Pa., secretary of the American Electrochemical Society, a list of members of this Society resident in the Pittsburg district, who are to be specially invited to the convention to be held at Pittsburg, May 5–7. C. E. Foster, Mem.Am.Soc.M.E., and John Brashear, Hon.Mem.Am.Soc.M.E., are among the speakers. Further announcement will be found on a later page of The Journal.

NECROLOGY

WILLIAM WILBERFORCE CHURCHILL

William Wilberforce Churchill died at Oshkosh, Wis., on March 24, 1910. He was born at Monroe, Wis., January 6, 1867, and was the son of Norman and Dr. Ann Sherman Churchill. After graduation from the Monroe High School in 1883, he spent one year at Rose Polytechnic Institute, and in 1886 entered Cornell University, from which he was graduated in 1889 with the degree of M.E. He was made a Fellow of Sibley College for 1889–1890, and received the degree of M.M.E. in 1890.

After graduation Mr. Churchill spent a few months with E. P. Allis & Co., Milwaukee, Wis. In 1890 he entered the employ of Westinghouse, Church, Kerr & Co., where he remained until his retirement because of a breakdown in health, in 1906. He rose through various intermediate positions in Chicago, Pittsburg and Boston to be chief mechanical engineer of the company's headquarters in New York. At the time of his retirement he was vice-president and director in the company. During his sixteen years of service he superintended the construction of the Boston terminal; the Kingsbridge power house, New York City; the Atlanta water plant, Georgia; the Lackawanna & Wyoming Valley R. R., Pennsylvania; The Grand Rapids, Grand Haven & Muskegon R.R.; Hotel Pontchartrain, Detroit, Mich.; the Northern Colorado Power Company, Denver, Colo.; the electrification of the Long Island R. R., New York, and many others. In 1902 he spent some time in Europe in connection with the electrification of the London Underground Railway.

Mr. Churchill was a member of the New York Railroad Club, the Cornell University Club, New York, the American Association for the Advancement of Science, and several Masonic orders.

GARDINER C. SIMS

Gardiner C. Sims, president of the William A. Harris Steam Engine Company, died at his home in Providence, R. I., on March 20, 1910. Mr. Sims was born in Niagara Falls, N. Y., July 31, 1845, and was educated there in the public schools. He began his engineering career with a four years' apprenticeship at the locomotive works of the N. Y. C. & H. R. R. R., West Albany, N. Y., afterward entering the navy yard at Brooklyn, N. Y., but returning to his former employers after three years, to become their chief draftsman. He next became

superintendent of the J. C. Hoadley Engine Works at Lawrence, Mass. Here he met Pardon Armington, with whom he formed a partnership for the manufacture of steam engines, both men devoting their ntire time to experimental work as a result of which they gave to the world the quick-running engine, in opposition to the established engineering practice and precedents. They built the first successful engine for Thomas A. Edison, which was sent to the Paris Exposition with his first dynamo, in 1881.

In 1876 Mr. Sims spent eight months at the Centennial Exposition and was appointed Democratic Commissioner from the State of Rhode Island to the World's Columbian Exposition in 1892, where he was made chairman of the Exposition committee on electricity, electric and pneumatic appliances, and was a member of the committee on machinery and transportation.

At the outbreak of the war with Spain, Mr. Sims volunteered, and was appointed Chief Engineer by the Navy Department and ordered to the navy yard at Boston. For his work in this branch of the service Mr. Sims was made a Lieutenant Commander and received congratulatory letters from Secretary Long and Engineer-in-Chief George W. Melville.

At the close of the war he was summoned by the War Department to assume the position of Superintendent Engineer of the United States Army Transport Service, and discharged his duties with honor until the completion of the work. He was appointed police commissioner in 1902, and at the time of his death was connected with the William A. Harris Steam Engine Company.

HARRY S. HASKINS

Harry S. Haskins, Associate Member of the Society, died at his home in Philadelphia on March 13, 1910. Mr. Haskins was born in Moretown, Vt., March 5, 1834, and at the age of twelve entered the machinists trade, first with Edwin Harrington and later with the Junction Shop, both in Worcester, Mass., where his family had moved. When Mr. Harrington went to Philadelphia, to engage in the building of machine tools, Mr. Haskins accompanied him, and soon afterwards the partnership of Harrington & Haskins was formed, which later became the firm of Edwin Harrington, Sons & Co. On the death of Mr. Harrington, the business became incorporated, with Mr. Haskins as president, an office which he retained until the time of his retirement, in 1900. Throug his mechanical ability and inventive faculty he added many improvements to the gear-cutting machines, hoists and overhanging railways manufactured by the firm.

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OF

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CONTENTS

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THE JOURNAL

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 32 JUNE 1910 NUMBER 6

SPRING MEETING, ATLANTIC CITY, MAY 31-JUNE 3

PROGRAM

Tuesday afternoon and evening, May 31

Informal reunion of members in the parlors of the Marlborough-Blenheim.

Wednesday, June 1, 10 a.m.

PROFESSIONAL SESSION

Business meeting. Reports of Committees, Tellers of Election, New Business.

PAPERS ON MACHINE CONSTRUCTION AND OPERATION

A Comparison of Lathe Headstock Characteristics, Prof. Walter Rautenstrauch.

The Strength of Punch and Riveter Frames made of Cast Iron, Prof. A. L. Jenkins.

Improved Methods in Finishing Staybolts and Straight and Taper Bolts as Used in Locomotives, C. K. Lassiter.

The Shockless Jarring Machine, Wilfred Lewis.

Wednesday afternoon and evening

The afternoon is left unassigned to give opportunity for sight-seeing. Roller chairs for the board walk will be available for the visiting members and guests through the courtesy of the Local Committee.

In the evening, entertainment on the Steel Pier has been provided by the committee.

Thursday, June 2, 10 a. m.

GAS POWER SECTION

BUSINESS MEETING AND REPORTS OF COMMITTEES.

PAPERS

A Regenerator Cycle for Gas Engines Using Sub-Adiabatic Expansion, Prof. A. J. Frith.

Gas Engines for Driving Alternating Current Generators, H. G. Reist.

Two Proposed Units of Power, Prof. Wm. T. Magruder.

Operating Experiences with a Blast Furnace Gas Power Plant, H. J. Freyn.

Thursday, 2 p. m.

PROFESSIONAL SESSION

MISCELLANEOUS PAPERS

The Mechanical Engineer and the Textile Industry, H. L. Gantt.

The Elastic Limit of Manganese and other Bronzes, J. A. Capp.

The Hydrostatic Chord, R. D. Johnson.

The Resistance of Freight Trains, Prof. Edw. C. Schmidt.

Thursday, 9 p. m.

Reception, followed by conferring of Honorary Membership on Rear-Admiral George W. Melville, U. S. N., Ret. A brief address will be made by Admiral Melville, and the evening will conclude with dancing and refreshments.

Friday, June 3, 10 a. m.

PROFESSIONAL SESSION

PAPERS ON POWER TRANSMISSION

Improvements in Lineshaft Hangers and Bearings, Henry Hess.

Experimental Analysis of a Friction Clutch-Coupling. Prof. Wm. T. Magruder.

An Improved Absorption Dynamometer, Prof. C. M. Garland.

Critical Speed Calculation, S. H. Weaver.

For several years the Spring Meeting of the Society has been held in cities where there has been an opportunity of visiting places of interest and inspecting engineering enterprises, and the time of the members has been very fully occupied in taking advantage of such excursions as the generosity and coöperation of the local members have made possible. While these meetings have all been thoroughly enjoyed, it was thought that it would be a welcome change to hold a meeting at a resort where those attending would have more time for renewal of acquaintance and for personal intercourse, instead of devoting so much attention to matters outside of the interests directly related to the Society and its membership. The last meeting of this sort was the Spring Meeting in 1903 held at Saratoga. The present meeting at Atlantic City should be an equally pleasant occasion, since there is no place in the country better adapted for holding a convention and the meeting is held at a time which is one of the most delightful in which to spend a few days on the New Jersey shore.

The headquarters of the meeting will be the Marlborough-Blenheim, situated at the central point of the board walk. The registration office will be located in the hotel, on the board walk floor, and the professional sessions will be held in the solarium on the office floor.

The Local Committee, chosen from the vicinity of Philadelphia, under the chairmanship of James M. Dodge, Past-President, Am.Soc. M.E., and with Arthur C. Jackson as secretary, is making provision for the entertainment. Through the courtesy of this committee, roller chairs on the boardwalk will be available and the use of the golf course at Pleasantville has been secured. Admission to the piers will also be arranged.

A Ladies' Committee has been formed with Mrs. Charles Day as chairman and will be in attendance at the headquarters to receive the visiting ladies and make their stay in Atlantic City most pleasant. A Bureau of Information will also be maintained.

LOCAL COMMITTEE

REPRESENTATIVES OF THE PROFESSION IN PENNSYLVANIA AND NEW JERSEY

JAMES M. DODGE, *Chairman* ARTHUR C. JACKSON, *Secretary.*

J. Sellers Bancroft
John Birkinbine
J. C. Brooks
Wm. C. Burnham
Harry W. Champion
James Christie
Walton Clark
Morris L. Cooke
Charles Day
J. J. DeKinder
Kern Dodge
Francis H. Easby
Peter Ehlers
Theo. N. Ely
Thomas M. Eynon
Stanley G. Flagg, Jr.
John Fritz
Harris R. Greene
G. T. Gwilliam
E. P. Haines
Robert E. Hall
Jas. T. Halsey
Henry Hess
Edward I. H. Howell
William C. Kerr
Wilfred Lewis
E. P. Linch
Chas. Longstreth
Thomas C. McBride
Chas. E. Machold
D. T. MacLeod
Edgar Marburg
Geo. W. Melville
Edwin A. Moore
Henry G. Morris
John S. Muckle
John C. Parker
F. R. Pleasonton
A. H. Ridell
T. F. Salter
Otto W. Schaum
Coleman Sellers, Jr.
Oberlin Smith
H. W. Spangler
A. A. Stevenson
Fred .W. Taylor
Geo. E. Titcomb
J. A. C. L. deTrempo
Wm. S. Twining
Harold Van Duzee
Mr. Van Gilder
S. M. Vauclain
William R. Webster
Tilden White
Walter Wood
J. E. Zimmerman

MEETING IN ENGLAND

A letter of detailed information covering all matters relating to the English meeting has been issued to all who have signified their intention of being in attendance there and particulars can be secured from the Secretary on request, by all who may be interested.

The party leaving New York on the official steamship Celtic, Saturday, July 16, at 2 p.m., will arrive in Liverpool Monday morning, July 25. Passengers will have a few hours for recreation on shore and will leave at noon by special train, for Birmingham, where they will be officially received by the Institution of Mechanical Engineers.

The Committee of the Society in charge of the arrangements is composed of Ambrose Swasey, *Chairman*, Charles Whiting Baker, *Vice-Chairman*, W. F. M. Goss, Geo. M. Brill, John R. Freeman, and George Westinghouse, Wm. H. Wiley, F. R. Hutton, Willis E. Hall, Calvin W. Rice, *ex-officio*.

PROVISIONAL OUTLINE OF MEETINGS

BIRMINGHAM PROGRAM

Monday, July 25

Afternoon.—Arrival in Birmingham.

Tuesday, July 26

Morning.—The Right Hon. the Lord Mayor of Birmingham and the Members of the Local Committee will receive and welcome the President, GEORGE WESTINGHOUSE, Esq., and the Officers and Members of the American Society of Mechanical Engineers, and the President, JOHN A. F. ASPINWALL, Esq., and the Council and Members of the Institution of Mechanical Engineers.

READING AND DISCUSSION OF PAPERS.

LUNCHEON in the Town Hall.

Afternoon.—Visits to Stratford-on-Avon, Worcester, Gloucester, or Bournville; and local Works.

Evening.—Garden Fête.

Wednesday, July 27

Morning.—Reading and Discussion of Papers.
Luncheon in the Town Hall.
Afternoon.—Visits to the University and local Works.
Evening.—Reception in the Council House, by invitation of the Right Hon. the Lord Mayor of Birmingham.

Thursday, July 28

Visits to Works in Coventry and Rugby; also to Warwick, Leamington, Kenilworth, or Lichfield.

LONDON PROGRAM

Thursday, July 28

Afternoon.—Arrival in London.
Evening.—Conversazione at the Institution.

Friday, July 29

Morning.—Papers on Electrification of Railways.
Afternoon.—Garden Parties at private houses.
Evening.—Institution Dinner in the Connaught Rooms, Freemason's Hall, Great Queen Street, W. C. (Including Ladies.)

Saturday, July 30

Morning and Afternoon.—Excursion by rail and river to Windsor and Henley (by invitation).
Evening.—Reception at the Garden Club in the Japan-British Exhibition at the White City.

SOCIETY NOTES

MEETING IN ST. LOUIS, MAY 27

The monthly meeting of the Society in St. Louis, originally announced for May 14, is to be held on May 27 at the Engineers' Club of St. Louis. The paper by Prof. Edw. C. Schmidt on Freight Train Resistance, published in The Journal for May, will be presented.

BOSTON MEETING, APRIL 27

At the meeting of the Society in Boston, April 27, in which the Boston Section of the American Institute of Electrical Engineers and the Boston Society of Civil Engineers coöperated, the paper on The Testing of Water Wheels after Installation, by Prof. C. M. Allen, Mem.Am.Soc.M.E., was presented by the author. The paper was published in the April number of the Journal. The speaker was introduced by Prof. I. N. Hollis, Chairman of the Boston Committee. Slides were shown in connection with the paper, and there was discussion by R. A. Hale, of the Essex Water Power Company, John C. Parker, Mem.Am.Soc.M.E., Prof. Dwight Porter, Prof. George E. Russell, and Prof. H. K. Barrows, of the Massachusetts Institute of Technology, Henry D. Jackson of Boston, and Henry C. Daggett. Mem.Am.Soc.M.E.

ENGINEERS LICENSE COMMISSION

The introduction in the New York Assembly of a bill requiring certain classes of practising engineers to pass examinations and have licenses as a condition of practicing their profession, led the Council to appoint a special committee to investigate this matter and advise what action, if any, the Society should take. At a meeting of this committee on April 27, resolutions were adopted expressing the opinion that any legislation affecting the rights and status of engineers could be most wisely originated by conferences between legislators and representatives of the national engineering societies. It was also voted to request those having in charge this legislation, to postpone action until after the Spring Meeting of the Society at Atlantic City, and to have the subject brought up for

discussion there. The Council was also requested to appoint a committee of five to report upon any proposed legislation affecting the interests of members of the engineering profession.

PRESENTATION OF MEDAL TO DR. C. J. H. WOODBURY

The association medal of the National Association of Cotton Manufacturers was awarded to Dr. C. J. H. Woodbury, Mem.Am.Soc.M.E., at the annual meeting of the association in Boston, April 27 and 28, 1910, in recognition of his work on the Bibliography of the Cotton Manufacture, and other services for the betterment of the industry. This medal, which was established in 1899, is awarded under very broad conditions that may include the author of any paper, a designer, either in mill construction or equipment, or of a process either of manufacturing or of finishing cotton goods. It has been given outside of the membership, although a member has generally been the recipient.

CORRECTION IN TRANSACTIONS, VOL. 26

The Forcing Capacity of Fire Tube Boilers by F. W. Dean, Transactions, Vol. 26, p. 93, in table, Par. 6, the last six items refer to fire-tube boilers instead of water-tube boilers.

CORRECTION IN TRANSACTIONS, VOL. 29

Ball Bearings, by Henry Hess, Transactions, Vol. 29, p. 447, caption of Fig. 14, Relation of Compression and Load for the Three Tests, Item 2 should read:

"2 balls $\frac{5}{8}$-in. diameter and flat disc. Compression according to Hertz $\frac{\delta}{2} = 0.0000805 \sqrt[3]{P^2}$"

STUDENT BRANCHES OF THE SOCIETY

STANFORD UNIVERSITY

The Stanford Mechanical Engineering Society, affiliated with The American Society of Mechanical Engineers, held its regular meeting on April 6, when Prof. Harris J. Ryan, Mem.Am.Soc.M.E., gave an interesting talk on the Los Angeles aqueduct, outlining its course of construction and showing the possible utilization of available power sites. A business meeting was held on April 20.

UNIVERSITY OF MAINE

Two papers were presented at the meeting of the student branch of the University of Maine on April 27: The Electric Car Control, by G. B. Chapman, being in the main a description of the hand and air-brake systems as used on electric cars, and The Modern Locomotive, by C. G. Cummings (1910), briefly describing the historical development, and discussing the different types of modern locomotives in use at present.

PENNSYLVANIA STATE COLLEGE

A meeting of the Pennsylvania State College student branch was held on April 28 and three papers were presented, on Applications of Refrigeration and the Manufacture of Artificial Ice, by Guy W. Jacobs (1910). The Different Processes of Mechanical Refrigeration, by George O. Weddell (1910), and The Design of a Cold Storage Plant, by Wm. R. Mollison (1910).

DEATH OF WALTER CRAIG KERR

Announcement is made of the death of Walter Craig Kerr, May 8, 1910, at Rochester, Minn. An account of his life will appear in an early number of THE JOURNAL.

OPERATING EXPERIENCES WITH A BLAST FURNACE GAS POWER PLANT

By Heinrich J. Freyn

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THE

JOURNAL

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

CONTAINING
THE PROCEEDINGS

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